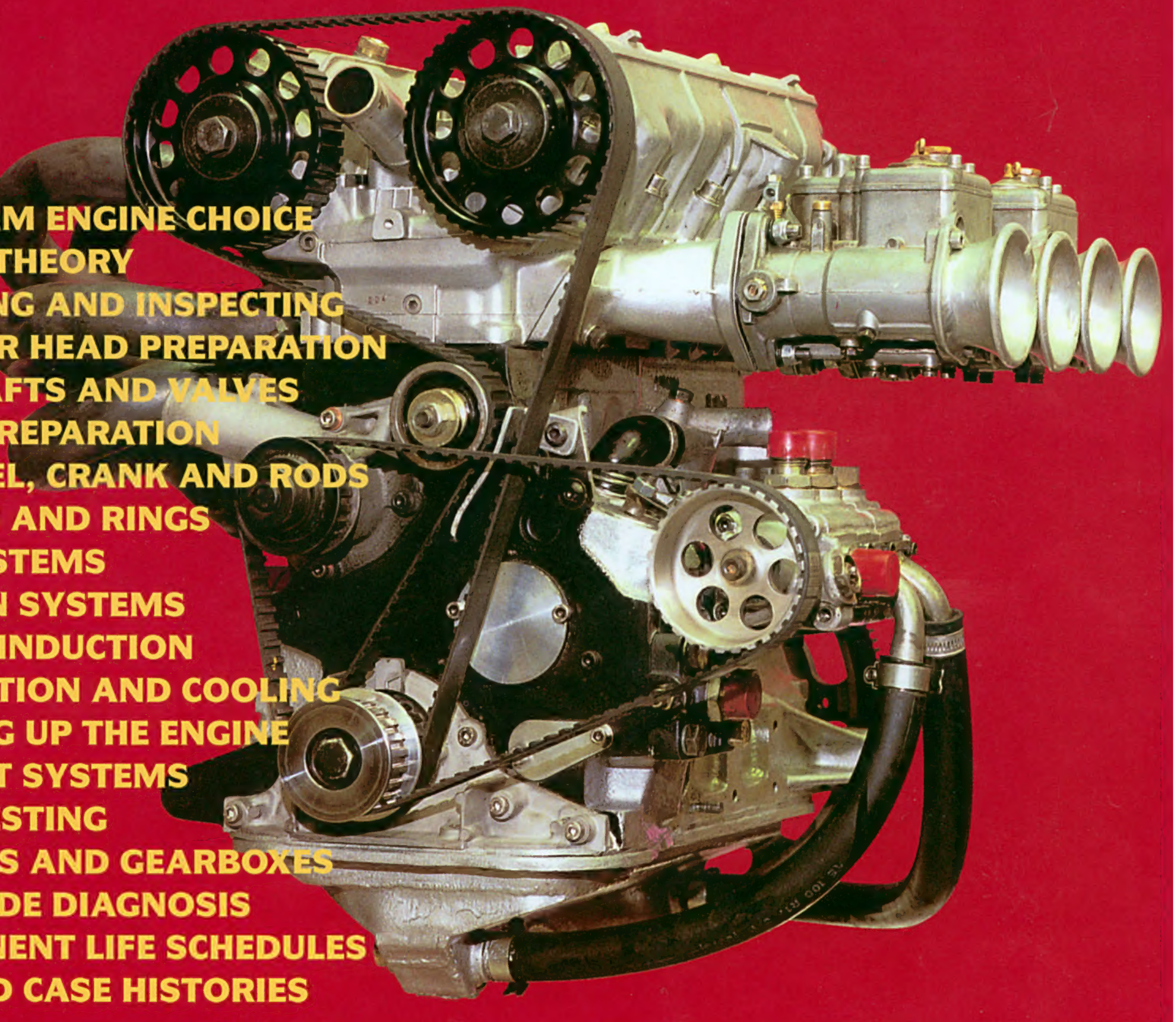
THE Guy CROFT WORKSHOP MANUAL

MODIFYING AND TUNING FIAT/LANCIA TWIN-CAM ENGINES



- TWIN-CAM ENGINE CHOICE
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Engine preparation expert GUY CROFT's Fiat and Lancia Twin-Cam engines are renowned across the world for their performance and reliability, having distinguished themselves in motorsport and on the road from the UK to New Zealand. Now, through the pages of this exhaustively detailed manual of engine modification, preparation and tuning, he has made available his years of experience at the sharp end of engine development to all users of Italy's most famous and versatile production engine. He also offers a clear explanation of the fundamentals of high-performance engine tuning, which will be invaluable to anyone seeking the ultimate from their car, whatever the source of its engine. The GUY CROFT WORKSHOP MANUAL is the essential reference source for the serious motorsport competitor.

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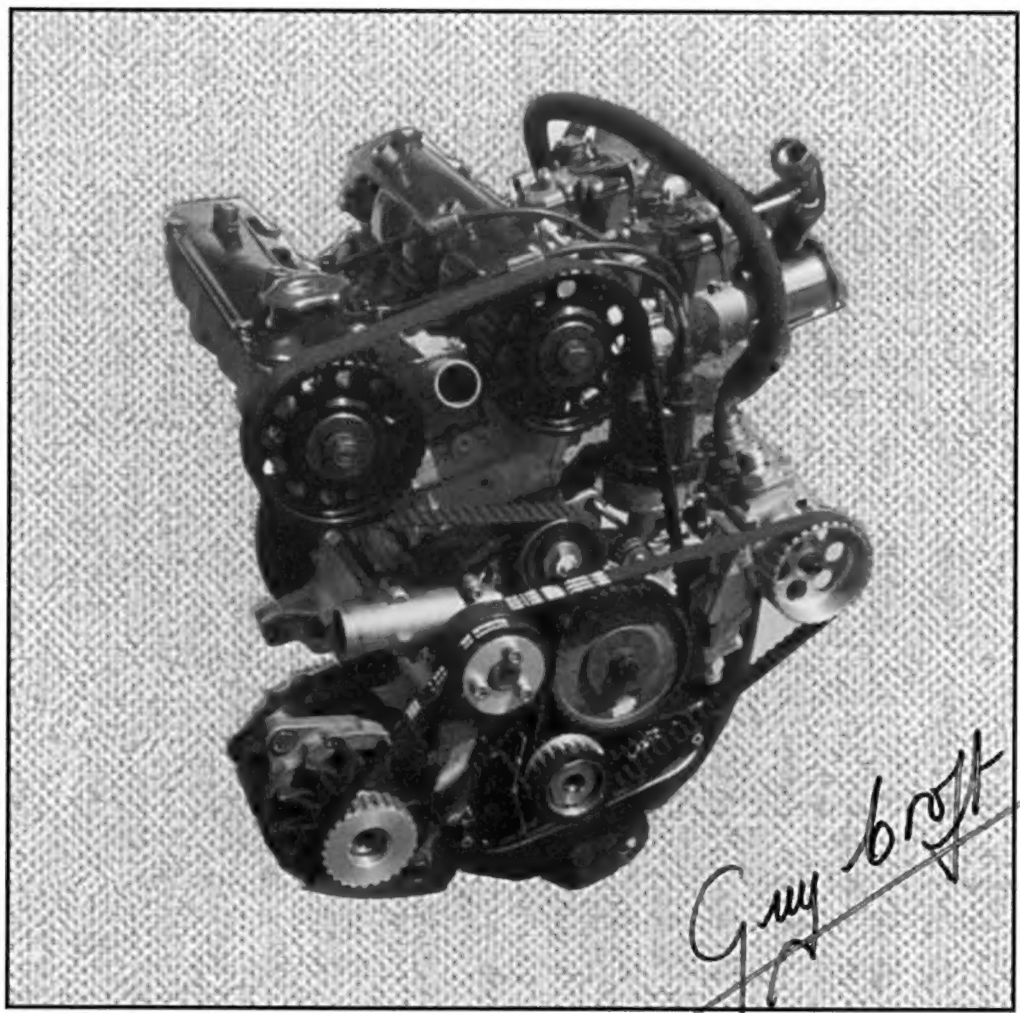
MODIFYING AND TUNING
FIAT/LANCIA TWIN-CAM ENGINES



Guy Croft

GUY CROFT BSc (Hons)

MOTOR RACING PUBLICATIONS

Introduction

Many years have passed since the forest and tarmac stages of the World Rally Championship echoed to the sound of the Fiat-Abarth 124 and 131 Twin Cams, but such was the success generated by these early cars, in which so many famous drivers made their name, that right up until the official withdrawal of Lancia from the series at the end of 1992, derivatives of those outstanding powerplants were still in use by the works teams. Importantly – as far as we are concerned – this heritage, spanning over two decades, continues to generate enormous interest among motorsport enthusiasts seeking a competitive engine for their clubman car.

This book is not about Abarth; for all their greatness, much of what they did was kept to themselves – perhaps understandably – so little information has ever appeared in print. Life would have been much easier if it had, and of course it would have made fascinating reading! Nearly all the techniques described in this book, therefore, have had to be derived from scratch.

The philosophy behind the work at Guy Croft Tuning over the last nine years has been quite simple – design the parts, test them thoroughly, then make them readily available at an affordable price. A great deal of hard work has gone into extracting the full potential from this remarkable series of engines, and despite the inevitable heartaches along the way, the progress made in establishing the TC as such a formidable and versatile high-performance powerplant has been a reward in itself. We hope that the work at GCT has made a genuine contribution to the recent resurgence of interest in these unique engines and to their popularity amongst both clubman competitors and high-performance road car enthusiasts.

To work in this field is particularly enjoyable. Fiat and Lancia Twin-Cam owners seem to share a common fascination for the intricate technicalities of special tuning; an increasing number of them have asked me: "Can you supply the parts if I build it myself?" My answer has always been "Yes", of course, but the purchase has so often been followed by in-depth, extensive telephone queries relating to the build-up and settings! It was in response to this that, two years ago, I discussed the possibility of this book with publisher John Blunsden, of Motor Racing Publications. John had been pleased with the success of *Fiat and Lancia Twin-Cams*, written by my friend Phil Ward, and to which I had contributed a chapter on special tuning; if there was a criticism, I was told by some readers, it was that 'my' chapter was not big enough.

My broad strategy has been to explain to the reader where to source a suitable engine, how to inspect and prepare it for road and competition use, and how to develop an integrated 'package' to eliminate running problems as far as possible in order that their precious budget is not wasted on an uncompetitive or unreliable engine. All the methods described are those actually used at GCT and are tried and tested. I hope readers will find of interest the technical justifications for the practical tuning methods described and will be suitably impressed by the section on owners' cars, the extent of which offers a clear indication of the wide variety of uses to which the TC can be put.

Acknowledgements

I would particularly like to thank those whose direct assistance in the preparation of this book was invaluable: Sue Mackinnon, manager of Medway Enterprise Centre, who willingly and cheerfully, in her spare time, converted my freehand scrawl into readable text over the course of 1995 – despite not having the vaguest understanding of the contents! My friend Ian Booth, photographer, who produced much of the specialist photographic material essential for a book of this type. Mike Hockley, who consistently supported GCT in every endeavour and whose incisive engineer's approach was a vital asset in bringing 'concepts' to practical reality. Gerard Sauer, for his submissions to the book, encouragement and assistance with queries on technical issues. Tom Casey, Irish Hot Rod Champion, who was the first driver supported by us to be able to supply the consistently accurate feedback needed to make rapid progress with development of a top race engine. Our involvement with Tom took us firmly, and for the first time, into 'super-tuning', where 'every little counts'. Tim Swadkin, racing engine contractor, who freely advised and assisted most generously during our dyno sessions at Warrior Automotive. John Woods, of Gemini Engineering, for his advice on crank balancing. Peter Newton and Giulietta Calabrese, of Fiat Auto (UK) PR, for their kind and persistent efforts to trace available photographs from Archivo Storico in Turin, and for their work in securing permission to reproduce valuable data from Fiat/Lancia workshop manuals. Additionally, GCT could not operate at all without the fullest backing of a first-class Fiat dealership's parts department, and in this respect I would like to record my sincere thanks to the staff at Montroe Motors, of Buckhurst Hill, Essex.

My gratitude, too, to the many companies who supplied important technical data on their products, all of which have been used very successfully by GCT. They include Titan Motorsport, Sachs Automotive, Datum, Fuel System Enterprises, Weber UK, Tran-X, ITG, Pacehigh, Pipercross, Lumenition, Canton-Mecca, Venolia Pistons and Warrior Automotive while *Cars & Car Conversions* magazine and the RAC Motorsports Association have also been most helpful in permitting us to reproduce valuable previously published material. Thanks also to the enthusiastic owners of Fiat/Lancia TCs who sent in photos and details of their cars, and the professional photographers whose work appears in the book. Finally, a special thank you to Richard Clark for his skill in converting such a complicated collection of tables, text and graphics into readable pages, and to John Blunsden for giving this book his fullest support.

Guy Croft
Keel Court
April 1996

Power/torque ratings

Fiat/Lancia bhp ratings are all expressed in DIN; in other words, the outputs are derived on engines fitted with water pump, alternator and silencer. GCT figures are not strictly DIN since in most cases a large-capacity, low-absorption silencer was used for dyno tests, and the alternator removed. Back-to-back tests indicate that for comparative purposes the GC figures may reasonably be assumed to be approximately 3% higher than the equivalent DIN rating.

Unless otherwise stated, all torque and bhp figures quoted are flywheel outputs, corrected to standard atmospheric conditions of humidity, pressure and temperature. This correction is important for comparison since an engine dyno-tested on a cold, damp day will have *corrected* outputs lower than the absolute figure and *vice-versa*.

CONTENTS

About the author

During the course of his 11-year service as an officer in the Royal Engineers, Guy Croft graduated from the Royal Military College of Science with an honours degree in automotive engineering. His interest in Fiat and Lancia Twin-Cams began in earnest in 1985, when the formation of Guy Croft Tuning gave him the opportunity to make a detailed appraisal of the works Group 4 Fiat Spider. His subsequent development work with these engines has given him a vast portfolio of experience, which has been widely sought after for numerous technical articles and publications.

Disclaimer

Whilst every care has been taken to ensure the correctness of the information in this book, all recommendations are made without any guarantee on the part of the author or publisher, who can accept no liability for loss, damage or injury, however caused, resulting from any advice given, or from errors in or omissions from the information provided.

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Front:
Completed just prior to publication, latest development GC St IV Fiat for NHRA oval racing. This engine represents the culmination of two years of top-level racing and around 60 hours of dyno-testing. Output was 156lbf ft @ 5750rpm, 200bhp @ 7500rpm, with not less than 146lbf ft torque between 4500–7000rpm. Despite having a flywheel weighing only 4kg, the engine ticked over quite happily at 600rpm and pulled full throttle/full load from 3000rpm. Specification includes: Titan 3-stage dry sump; GC IIID exh cam, IVB inlet (hybrid); 46/40 valves, triple springs, alloy caps, GC alloy verniers, 1" belt; 130 TC head gasket, race bolts, fully ported/blueprinted, 45DCOEs (40 chokes); GC/Venolia forged pistons, 11.5:1 CR (2049cc); production Bosch electronic ignition, NGK B9 EGV plugs; lightened crank, rods; F3 5½" clutch, steel flywheel. Engine was sensationally quick on its debut at Rose Green, Tipperary, and was raced consistently to 9000rpm.

Top rear:
When it comes to Fiat/Lancia conversions, nothing is necessarily what it seems. This '037' is in fact a shortened, highly modified Lancia Monte Carlo fitted with a St II 2/ GCT Lancia Twin Cam. Thousands of man-hours (and pounds!) went into building this stunning replica and sourcing original decals, wheels and other details.

Centre rear:
Outstanding National Hot Rod (NHRA) spaceframe raced by Irish Champion Tom Casey. Designed and built by well-known Ford rally specialists Autocross, of Bracknell, with over 20 years of motorsport experience behind them, this tubular-steel/Kevlar vehicle was sensationally quick on its debut. Only six months from prototype to race-ready, the first Autocross car won three races in succession 'out of the box'.

Lower rear:
St II 1600 Fiat installed in Westfield Eleven (see Chapter 18) developing around 145bhp and tuned for sprint/hillclimb events.

CHAPTER 1

TWIN CAM MODELS

Analysis of the engines most commonly used for motorsport

The purpose of this chapter is to identify the key characteristics of the TC engines most commonly used for motorsport and the vehicles in which they may be found. The list, inevitably, is by no means a complete compendium of every single model – that would fill a book by itself. Instead it is a guide for the clubman motorsport enthusiast who wants to know which engine will be suitable for his purpose, and is based on knowledge acquired over the years by Guy Croft Tuning (henceforward referred to as GCT) of the most popular and cost-effective units. While respecting the purist view, *ie* retention of the original engine with the car (and there are many such fine examples within the various Fiat/Lancia Owners' Clubs), it is true that the demise of numerous Fiat and Lancia cars has made available an enormous range of potential powerplants, which have design features so outstanding that their adoption as clubman competition engines is assured for a long time to come.

It may rightly be said that as such the Fiat/Lancia Twin Cam has no equal. With hindsight, it is easy to see that the design concept of these engines probably had motorsport in mind. Fiat's TC designs (later adopted for the Lancia range) dominated World Championship rallying in the Seventies. They were only really matched by the Ford/Cosworth BDA, which by comparison with the Fiat TC was produced in tiny numbers. The Fiat 131/132 2-litre engine, in particular, is one of the classic production engines of all time, and having been conceived with Gp 4 homologation in mind, was essentially a race engine fitted in a production car, a point lost on most owners more preoccupied with corrosion.

For example, few if any volume manufacturers went to the trouble of installing heat-treated forged crankshafts (capable of withstanding the punishing torque of the works 131 cars) in their production vehicles, and few other manufacturers have left such an outstanding fundamental design so little altered in over 20 years. Anyone who takes the trouble to compare the intervals of, say, a 131 2*l* and an Integrale 8v will see this for themselves. The Fiat/Lancia TC truly has an impeccable motorsport pedigree.

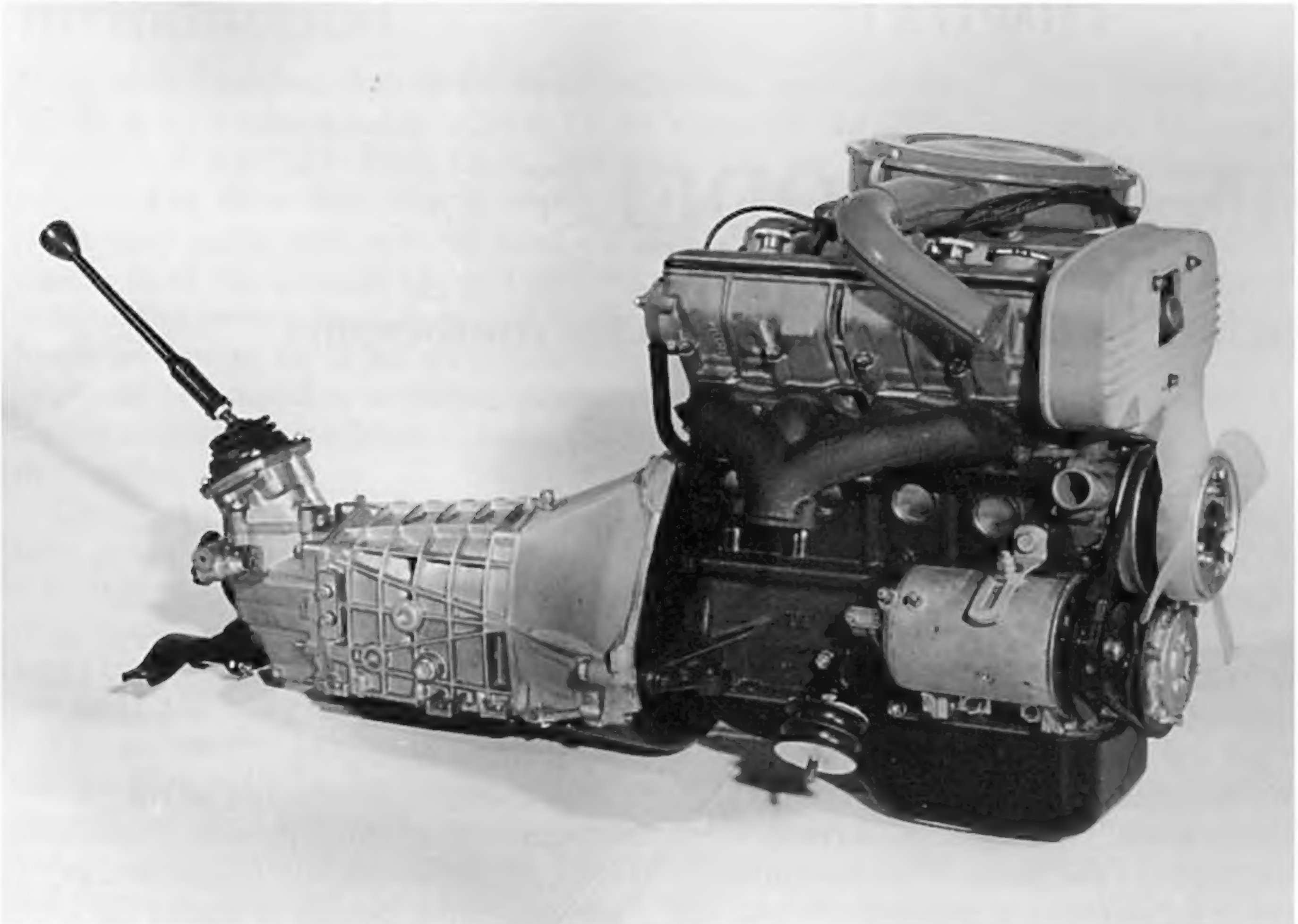
The spin-off from this for the clubman enthusiast is obvious: All the TCs are tried and tested. Design flaws, manufacturing defects and inherent problems are (as far as GCT has been concerned) unheard of. The potential of the entire range for uprating is phenomenal, and the ease with which this may be carried out is quite unique to the Fiat/Lancia TC. No other engine in its class can boast such a multiplicity of good points! The clubman can enjoy an enormous amount of fun with a low-budget 1600 in a modest kitcar, or go all the way and be competitive at National level with a fully prepared 2*l* – using in each case a donor engine bought readily for very low expenditure. Indeed, it is no exaggeration to say that the internals of many Guy Croft engines now winning races and setting records have already spent their former life as the powerplant in a passenger vehicle!

Selecting the key components and offering a grading system for the models listed is extremely hard. All the engines have virtues in their own right, and the ease with which components may be changed, or uprating modifications carried out, is common to all except, of course, that the 'mapped' engine (*ie* a late turbo model) presents its own peculiar problems of setting-up.

For example, irrespective of the standard valve, cam type and compression ratio, all the models will readily accept different cams, pistons of higher CR (normally aspirated versions) and, if the particular model will not accept larger valves on the standard seats, big valve seat inserts can even be fitted up to very large sizes. Other engines do not have this flexibility: the VW and Peugeot engines used extensively for National Hot Rod racing have valve sizes roughly comparable with standard TC items and the valve train layout will not permit bigger; the 2*l* Alfa Romeo is very restricted on valve size, cam lift and port size and lacks the Fiat/Lancia's torsional rigidity. Where the owner of another engine needs more cubic capacity and needs to 'restroke' the crank, the Fiat/Lancia TC owner simply moves from a 1585 to a 1756 or 1995cc model.

The main object of the 'marks out of

1/1: Fiat 124 Sport BC series.



1/2: Early 1608 (125) engine with electrostatic fan and centrifugal oil filter. Gearbox and bellhousing will not fit later 132 type engines, which used a larger flywheel. Box was a weak point anyway.

10' scoring system introduced in this book is to identify how much power can be obtained, reliably, from the least expenditure. In this respect, a steel-crank 1600 tends to score more than a cast iron-crank version for reliability reasons (though GCT have never seen a 1585 crank failure, even on a highly tuned Delta Turbo *ie*), and then again, a turbo 1600 may score more than a normally aspirated version because it inherently produces more bhp per £ spent. The reason for scoring the 2*l* normally aspirated versions highly against the 2*l* turbo engines is simply that turbos are usually classed with a cubic capacity equivalence factor of 1.4, taking them into a higher class, and therefore a direct comparison is difficult.

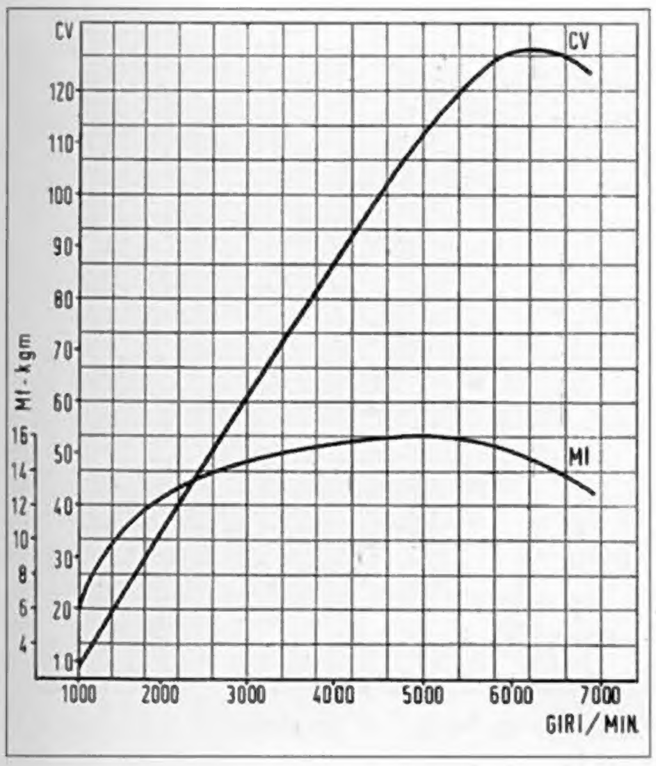
When choosing a powerplant for a road-only vehicle it is obvious that, given the extra expenditure required for the donor engine (plus ancillaries, ECU and harness), the turbo 2*l* versions deliver such blistering performance (because of their phenomenal torque) that they make an obvious choice – if they can be obtained in good condition! The 130 TC scores more than the 131 2*l* because it already comes with twin carbs, big inlet valves, a 4-2-1 tubular exhaust and oil cooler; on the other hand, there are few about.

Ultimately, readers who wish to extend themselves beyond the boundary of fitting a standard engine will realize that the choice of powerplant is really only governed by the size of the engine *vis a vis* their particular competition class, the question of normally aspirated/turbo, the budget they wish to commit to the project and, of course, the availability of a suitable donor engine in their locality. Specifications of engines from potential donor cars illustrated in this Chapter will be found on pages 13–15.

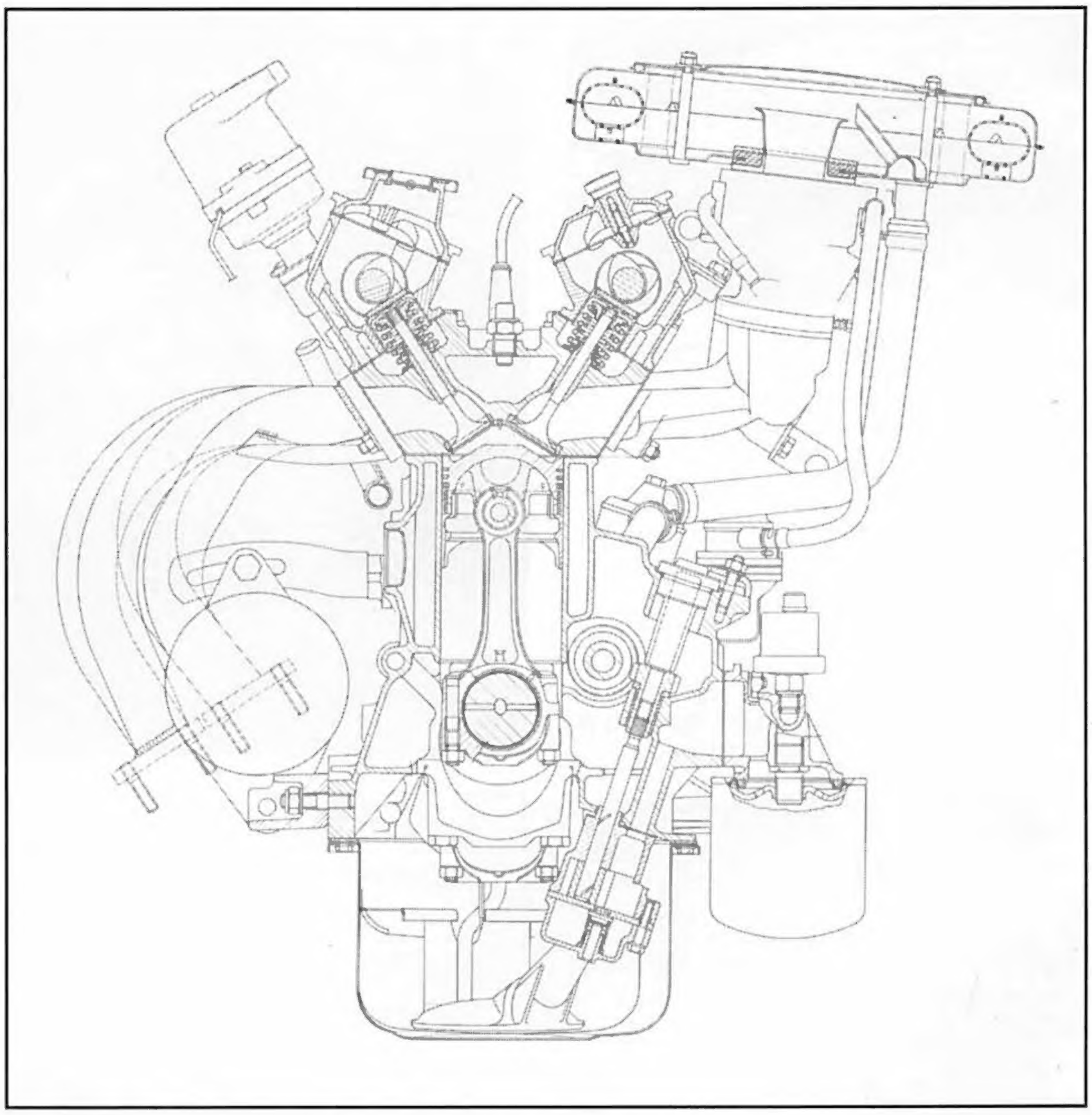
Further useful reference to and interesting reading about the wide variety of engines fitted to Fiat and Lancia cars can be found in *Fiat and Lancia Twin-Cams* (ISBN 0-947981-57-8), by Phil Ward, and *Lancia Beta: A Collector's Guide* (ISBN 0-947981-62-4), by Brian Long, both published by Motor Racing Publications Ltd.

1/3: 1756 Fiat Spider (CS version).

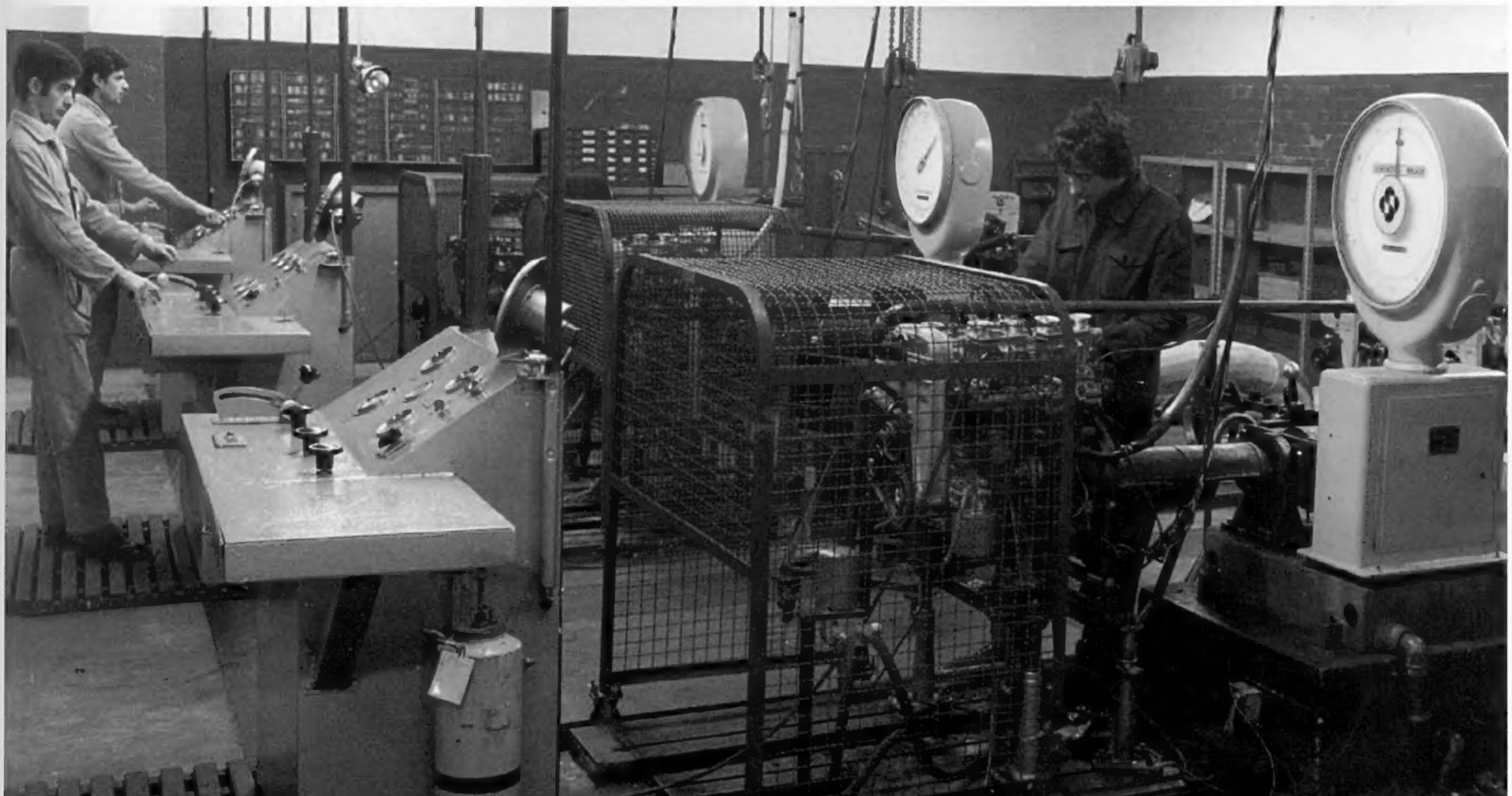
1/4: Author's own 124 Abarth Spider pictured in 1984.



1/5: Power curve for 124 Abarth Spider engine. (Fiat Auto SpA – copyright reserved)

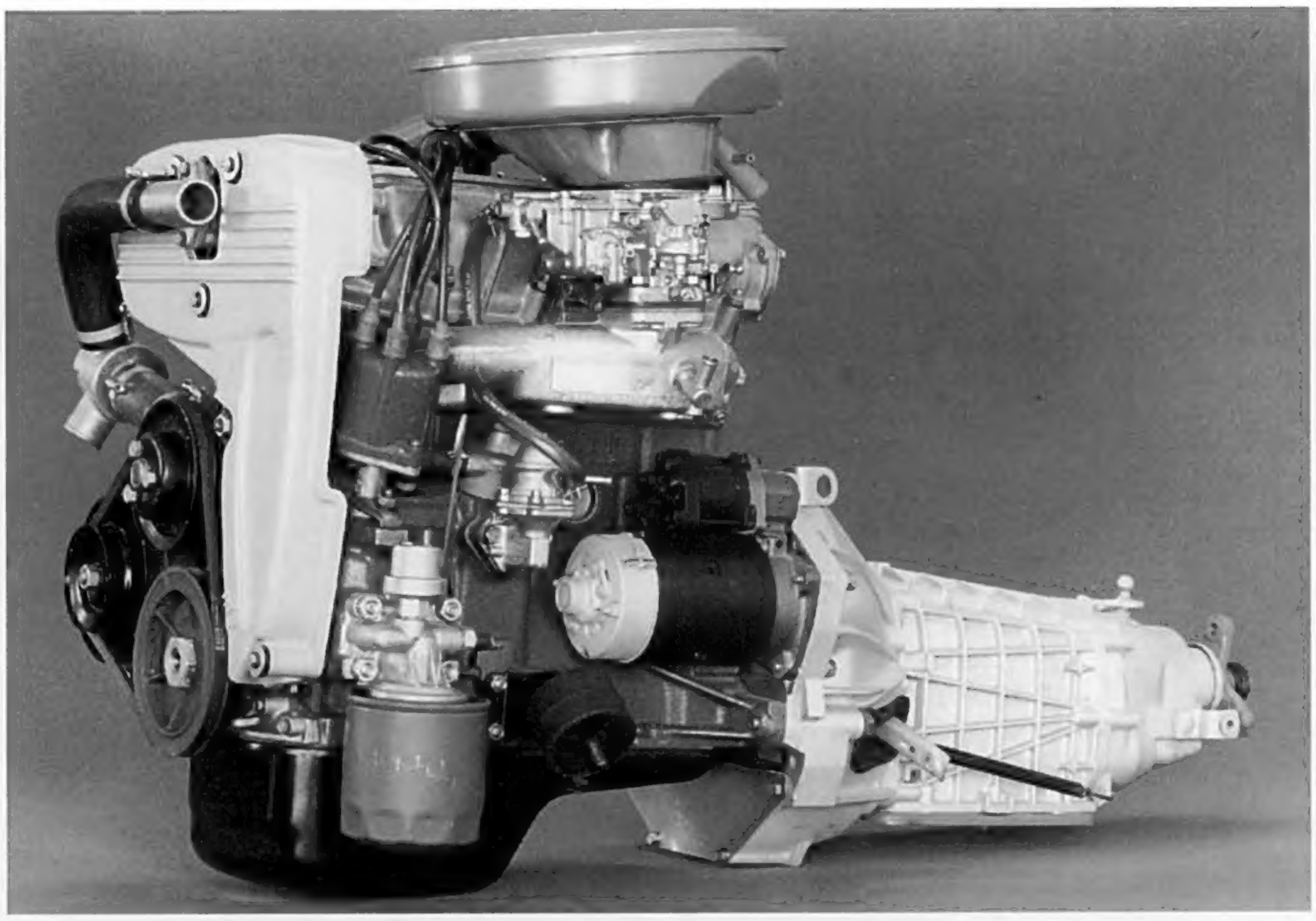


1/6: Extract from Fiat 124 Abarth 1800 manual. (Fiat Auto SpA – copyright reserved)

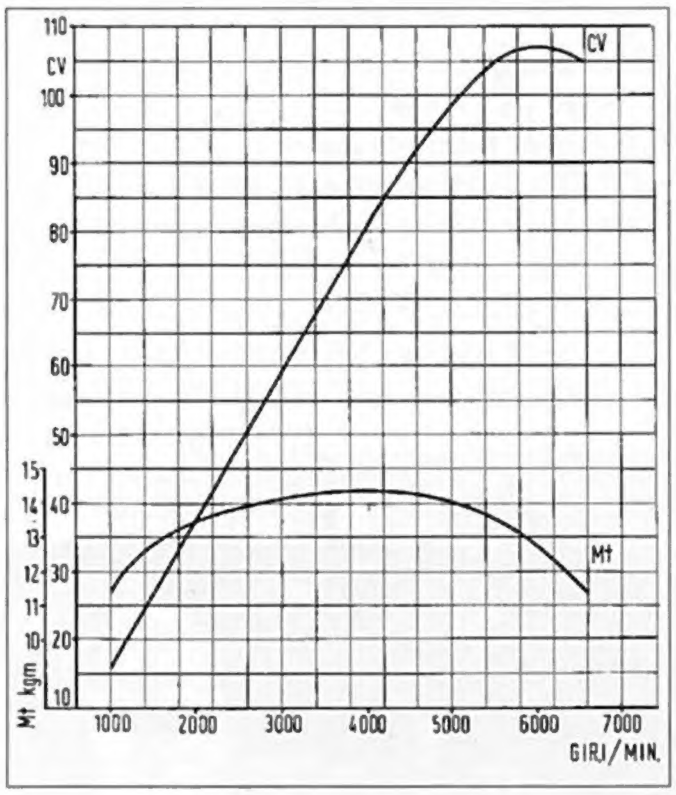


1/7: Rare factory shot of 124 Abarth Spider 8v engines on dyno at Abarth. Centre engine is under load. One of the better jobs in the world, but note the complete absence of soundproofing!

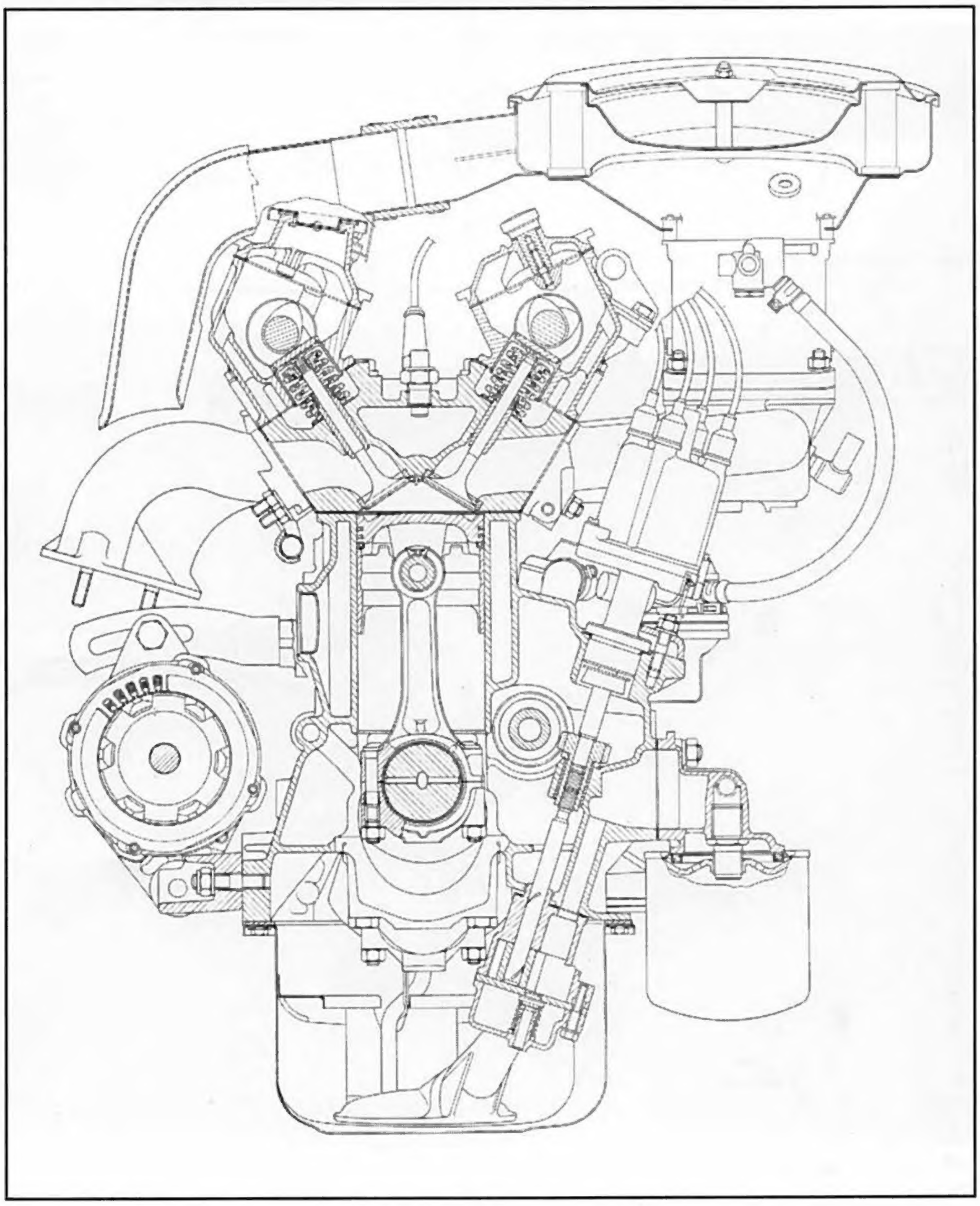
1/8: Fiat 132 (1592cc): boxy, heavy car, nice engine. Unfortunately most of these cars have been crushed. Also available with 1756 and 2l engines. 1592 shares same crank, rods with 1756, ie is in effect same engine but with smaller bores.



1/9: 132 1800 engine (right). Larger bellhousing for 220mm dia (friction face) flywheel, but again weak gearbox (140bhp max!).

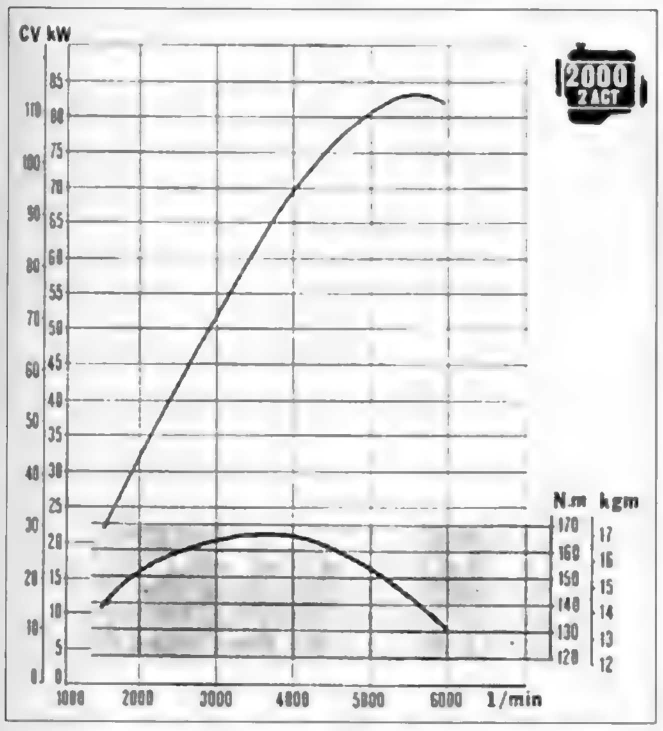


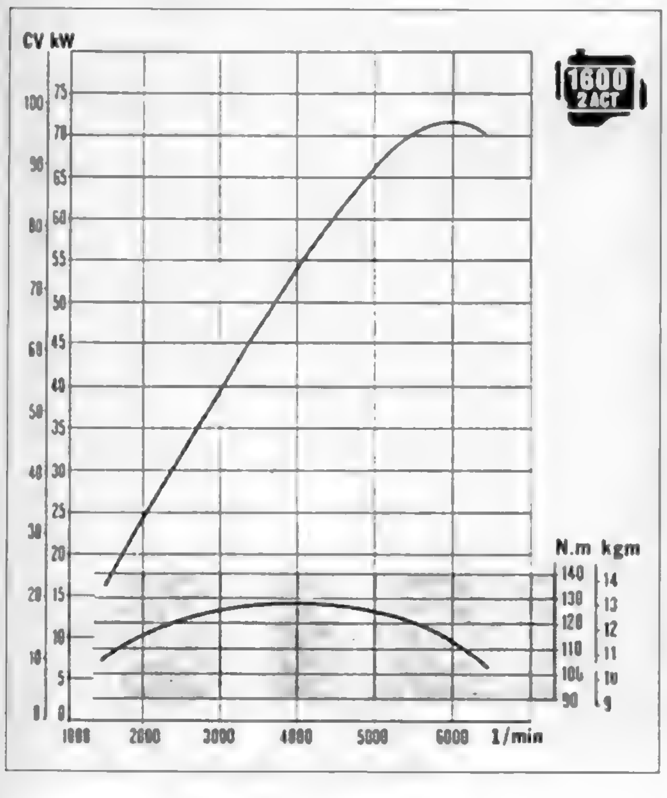
1/10: Power curves of 132 B1000 engine. (Fiat Auto SpA – copyright reserved)



1/11: Cross-section of 132 1800 engine, viewed from the crank nose. Fiat have always produced workshop manual drawings of superb quality. (Fiat Auto SpA – copyright reserved)

1/12: Pristine 131 Sport owned by Houghton, Cambridgeshire solicitor Anthony Fausset.



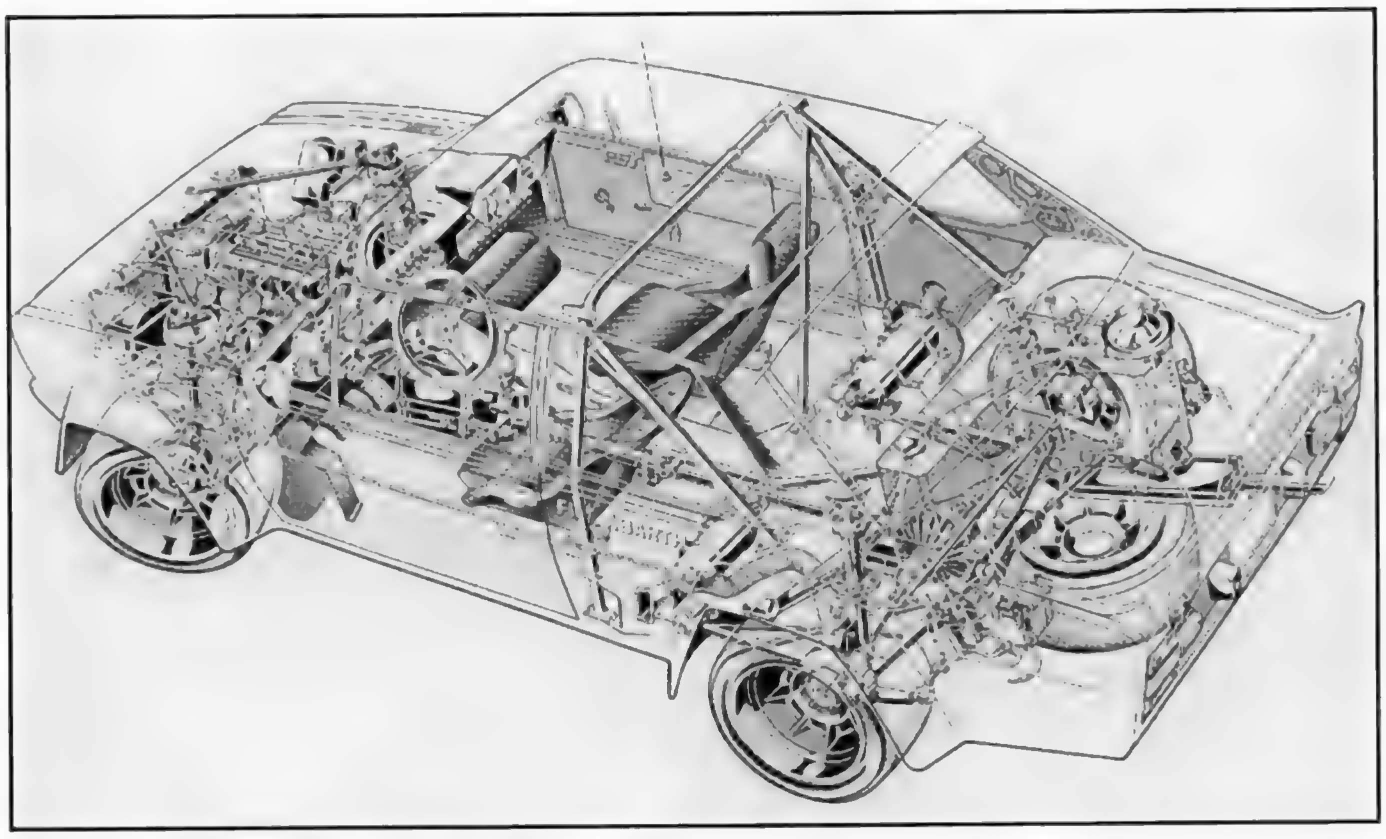


1/13: Power curves for the 2000 and 1600 engines reproduced from the Fiat 131 manual. (Fiat Auto SpA – copyright reserved)

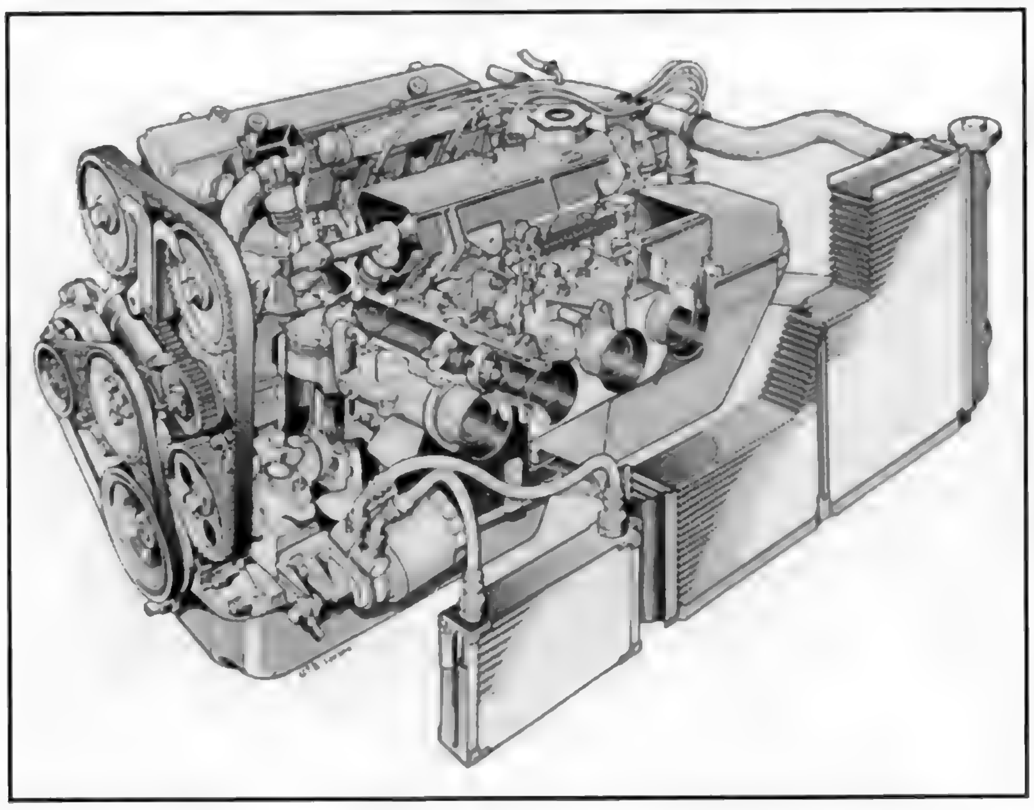
1/14: Promotional shot of two-door 131 Sport (known as 131 Racing in Europe).

1/15: High on the author's Christmas list – an original, works-prepared 131 Abarth Rally (Gp 4) – arch enemy of the Mk 2 Ford Escort in the '70s!

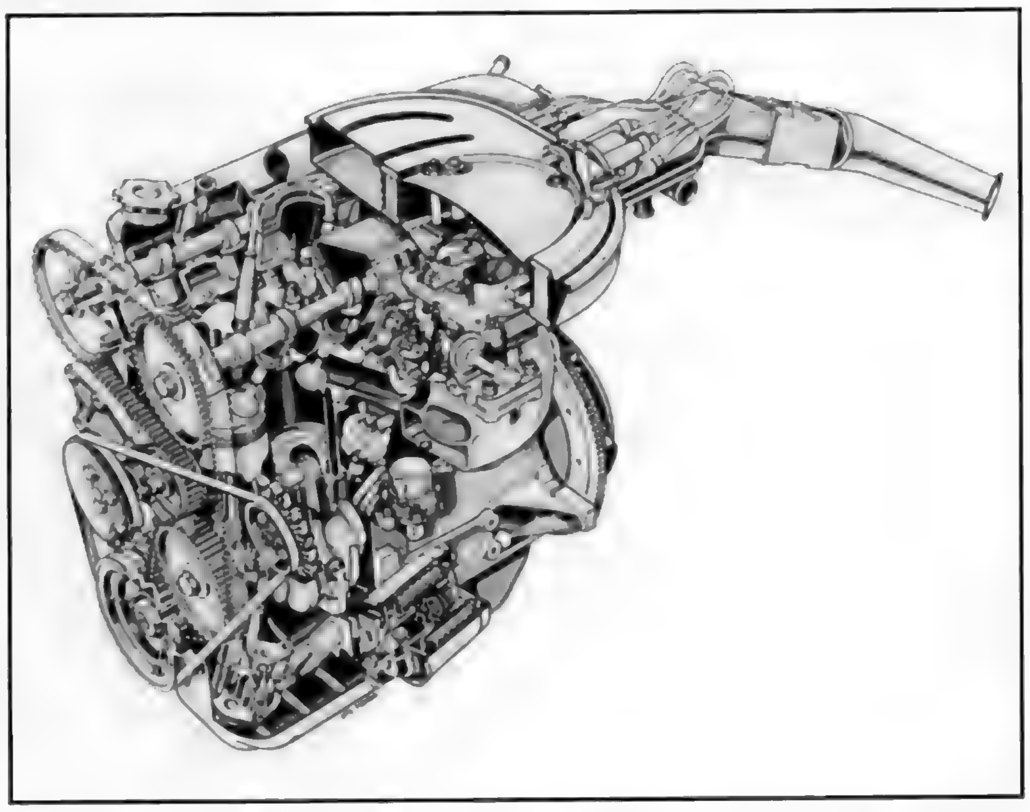
1/16: Preparing a clubman rally car? Cutaway of this works 131 Rally (16v, fuel-injected) shows the way to go...



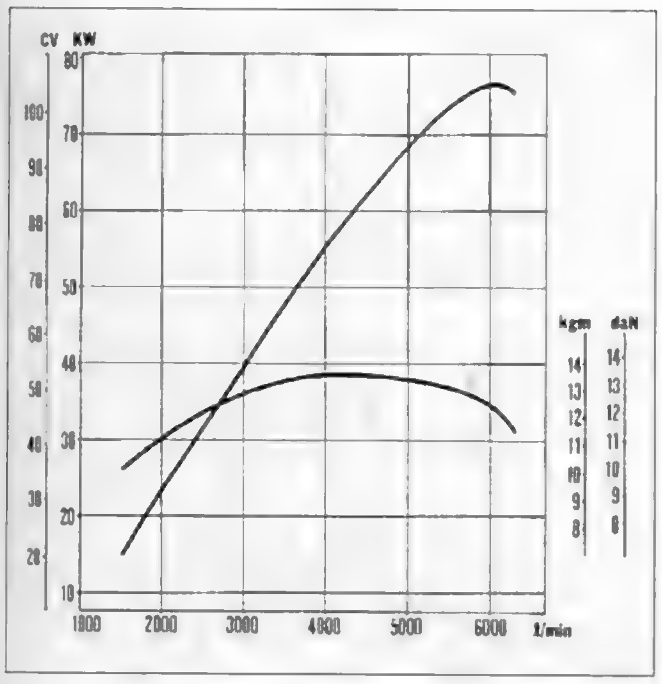
1/17: Fiat Strada Abarth 130 TC.

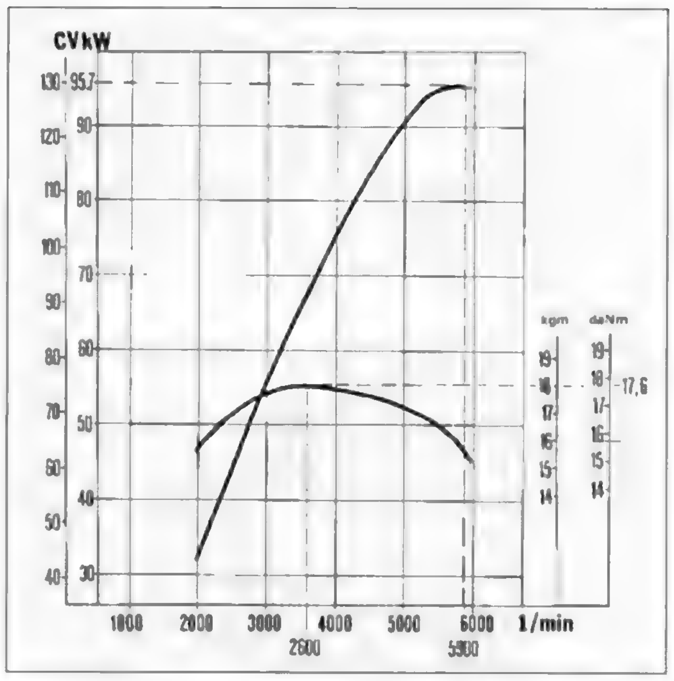


1/18: Fiat Strada Abarth 130 TC engine and ancillaries (one of the last production cars in the world to be produced with twin carbs).

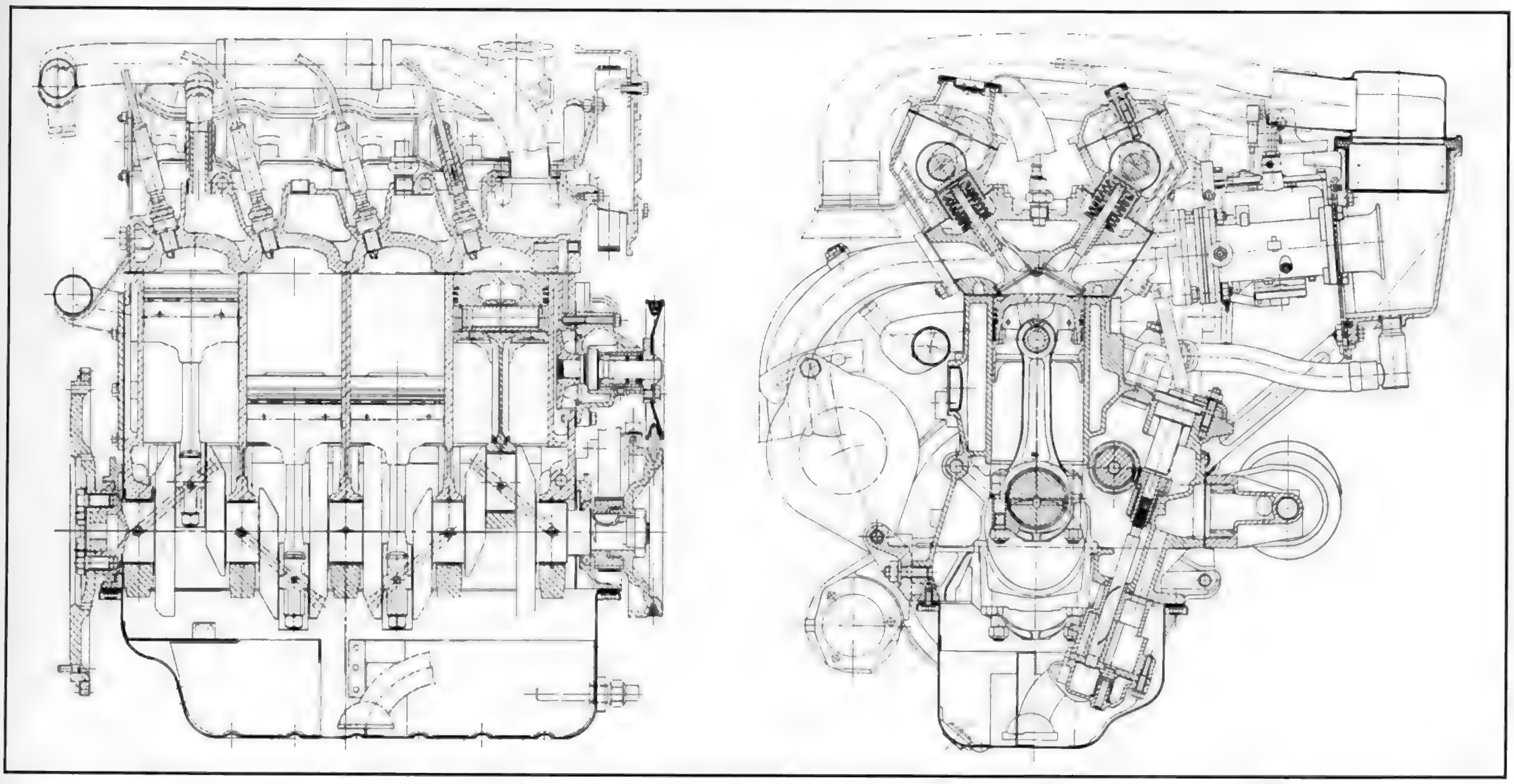


1/19: Regata 100S engine – similar to 105 TC, Delta/Prisma 1600 GT. Note oil pump design, also used on 130 TC and Delta Turbo 1.6 (carb).





1/20, 1/21: Engine power curves from the 105 and 130 TC workshop manuals. (Fiat Auto SpA – copyright reserved)

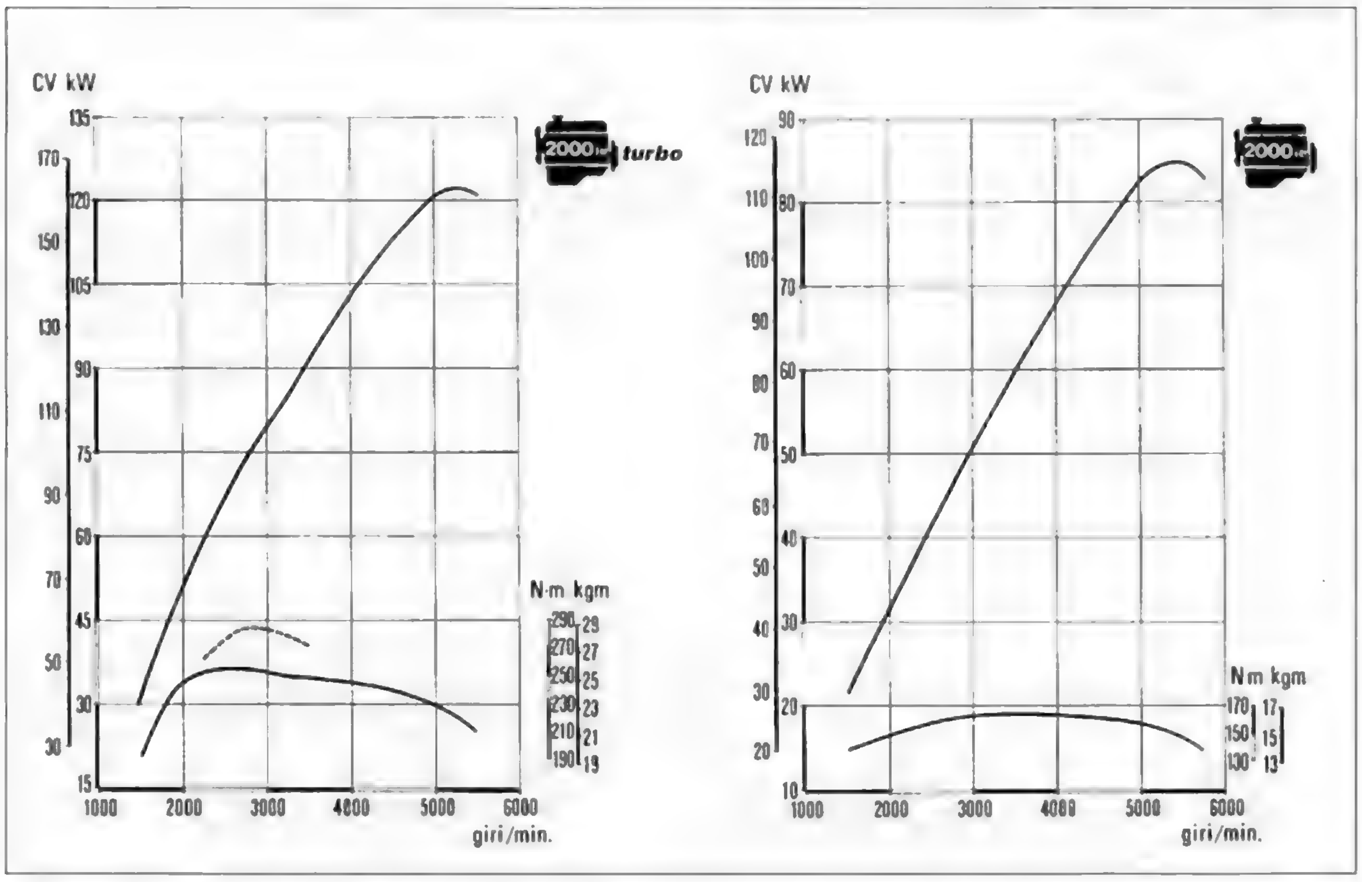


1/22: Cross-sections of 130 TC engine. Note slight upward tilt on carbs. 130 TC is only Fiat model since 124 BC 1608 to use twin carburettors, either Solex or Weber 40s. Distributor driven off inlet camshaft end. Note full-form transverse sump and oil pickup design. Sumps use a 'cruciform' baffle layout and are quite effective. Delta and Prisma versions use remote filter set-up. (Fiat Auto SpA – copyright reserved)

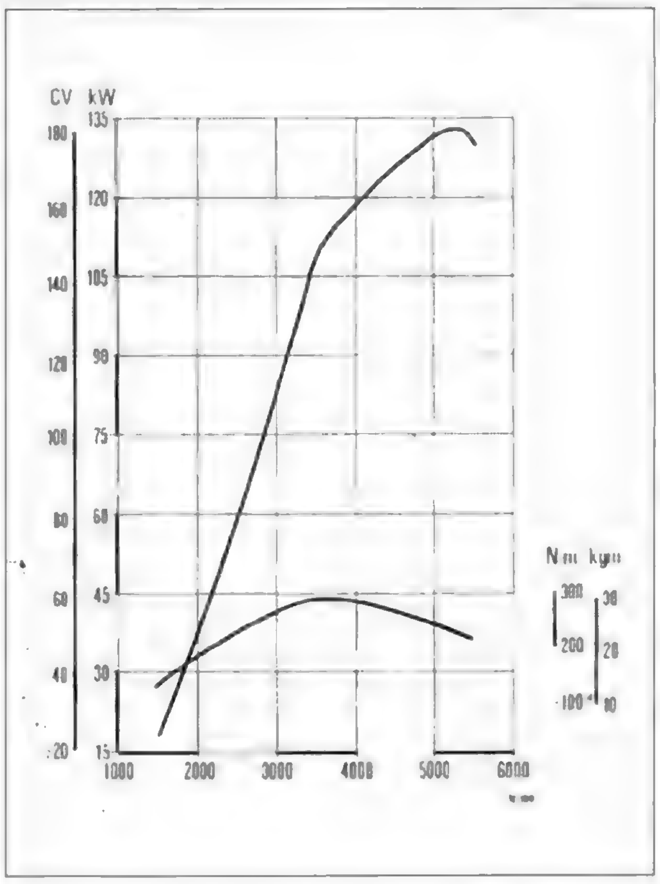
1/23: Lancia 2l Beta Spider owned by Tim Walker. A pristine example of this now rare model.

TWIN CAM MODELS

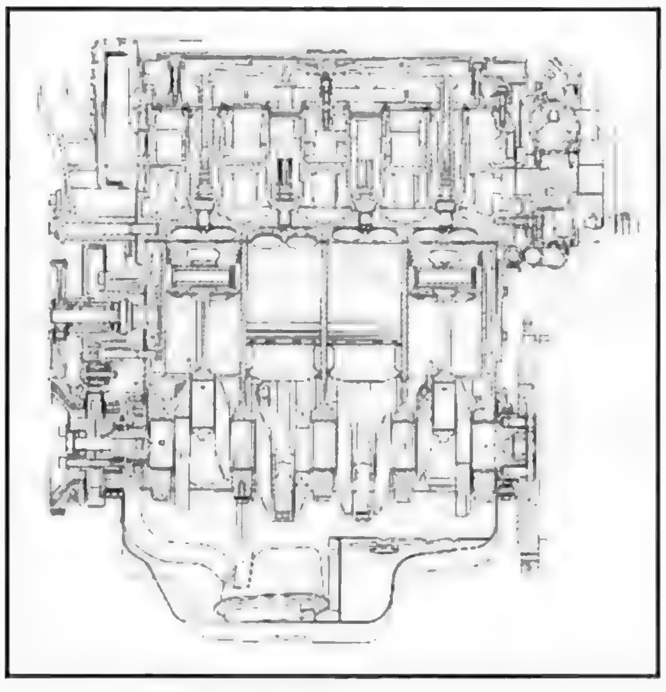
1/24: Works shot of a Gp A Lancia Integrale (16v). These cars dominated World Championship rallying for nine years.



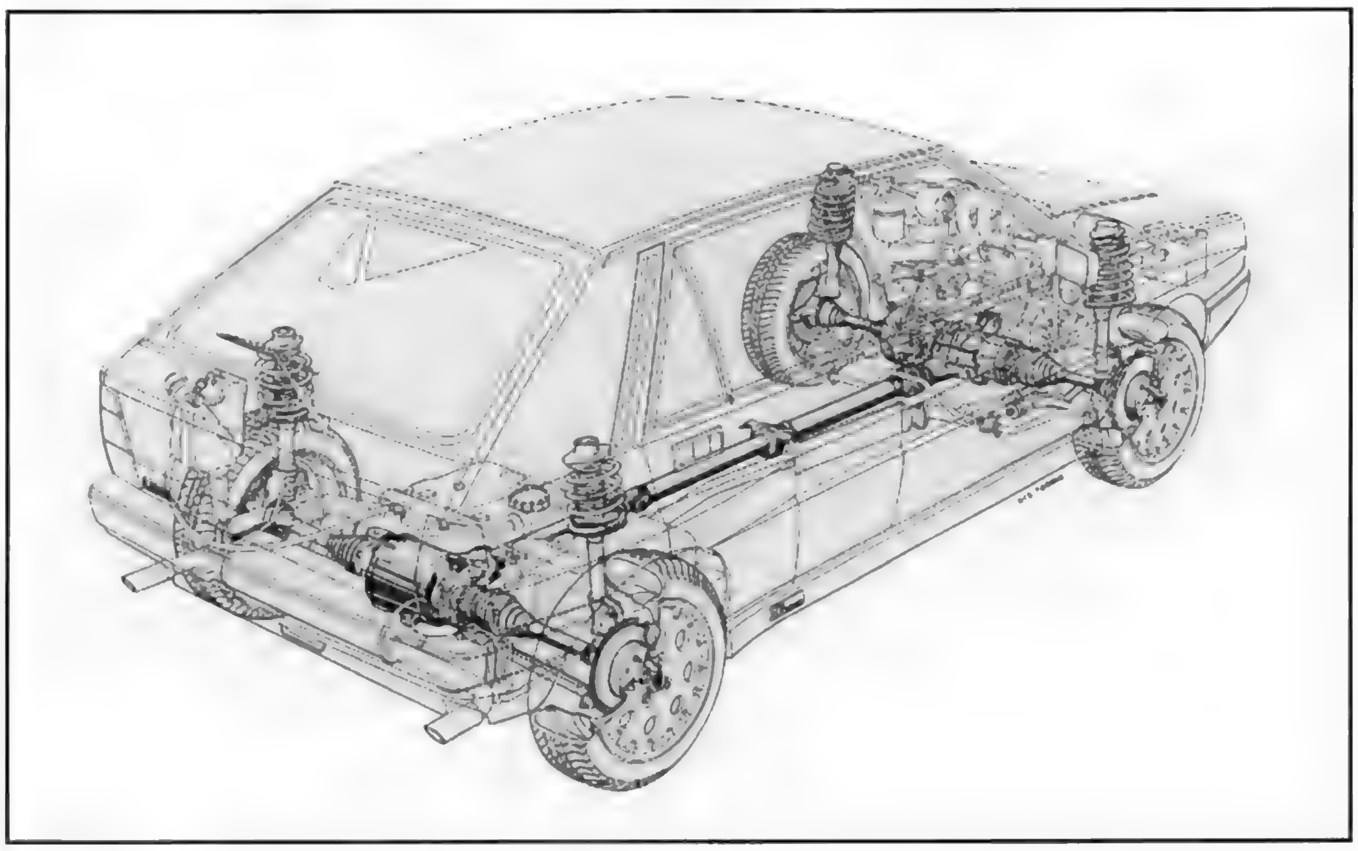
1/25: Power/torque outputs from Delta, Prisma 4WD versions (normally aspirated and turbo). (Fiat Auto SpA – copyright reserved)



1/26: HF Integrale outputs. (Fiat Auto SpA – copyright reserved)



1/27: The 16v 2000 ie engine was available as normally aspirated (eg Thema ie 16v) and turbo (Integrale, Thema turbo) (Fiat Auto SpA – copyright reserved)



1/28: Cutaway of HF Integrale (2l, 8v, turbo) showing drive-train layout. (Fiat Auto SpA – copyright reserved)

DONOR CARS AND ENGINES

Sources and specifications of the engines most commonly used for motorsport

124 SPORT COUPE/SPIDER ENGINE DATA

SERIES	BORE/ STROKE (mm)	CR	YEARS	CAPACITY (cc)	VALVES (mm)	CAM TIMING IN (deg)	EX	LIFT (mm)	POWER (bhp DIN)	TORQUE (lbf ft)
132AC000	80 x 79.2	9.8:1	73–77	1592	42.4/36	12/53	54/11	9.7	108 @ 6000	109 @ 3400
125BC000	80 x 80	9.8:1	70–73	1608	42.4/36	26/66	66/26	9.5	110 @ 6400	103 @ 3800
132AC100	84 x 79.2	9.8:1	73–77	1756	41.8/36	15/55	55/15	9.9	118 @ 6000	

Engine tuning notes:
All models have early con-rod design – race bearings not available. 1592 model has 12mm dia flywheel bolts, others have 10mm.
All models have forged steel crank.
Some 1608 models have exhaust cam-driven distributor, twin 40 IDF.
Parts now hard to get on early models, especially 80mm pistons.
Engine tuning potential 6/10.
Engines will accept later 131 gearbox and bellhousing (but 125 type needs larger 132 type flywheel).

124 ABARTH SPIDER ENGINE DATA

SERIES	BORE/ STROKE (mm)	CR	YEARS	CAPACITY (cc)	VALVES (mm)	CAM TIMING IN (deg)	EX	LIFT (mm)	POWER (bhp DIN)	TORQUE (lbf ft)
132AC4000	84 x 79.2	9.8:1	73–75	1756	41.8/36	15/55	55/15	9.9	128 @ 6200	121 @ 5000

Engine tuning notes:
Original engine essentially 124 Sport 1756, plus twin 44 IDF, 4-2-1 Abarth exhaust manifold system. Numerous Gp 4 parts fitted by owners over the years so original engines are extremely rare. All 8V versions had exhaust cam-driven distributor.
Engine tuning potential 6/10.
Later Gp 4 versions had Abarth 16v head – highly sought after!

FIAT 132 ENGINE DATA

SERIES	BORE/ STROKE (mm)	CR	YEARS	CAPACITY (cc)	VALVES (mm)	CAM TIMING IN (deg)	EX	LIFT (mm)	POWER (bhp DIN)	TORQUE (lbf ft)
132A1000	84 x 79.2	8.9:1	73–75	1756 (1800)	41.8/36	12/53	54/11	9.7	105 @ 6000	104 @ 4200
132B1000	84 x 79.2	8.9:1	75–77	1756	41.8/36	12/53	54/11	9.7	107 @ 6000	104 @ 4200
132C2000	84 x 90	8.9:1	–82	1995 (2*l*)	41.8/36	15/55	57/13	9.9	112 @ 5600	Around 116 @ 3000
132D1000 (Argenta)	84 x 90	9:1	83–85	1995	41.8/36	15/55	57/13	9.9	113 @ 5600	Around 116 @ 3000

Engine tuning notes:
All models have late con-rod design with 12mm dia flywheel bolts.
All models have same flywheel diameter.
All models have forged steel crank, 2*l* versions are Tufftrided/nitrided.
Late gearbox and bellhousing was introduced on 2*l* versions.
Engine tuning potential: 1800 – 7/10; 2*l* – 9/10.

TWIN CAM MODELS

FIAT 131 SUPERMIRAFIORI ENGINE DATA

SERIES	BORE/ STROKE (mm)	CR	YEARS	CAPACITY (cc)	VALVES (mm)	CAM TIMING (deg) IN	EX	LIFT (mm)	POWER (bhp DIN)	TORQUE (lbf ft)
131B1000	84 x 71.5	9:1	77–85	1585	41.8/36	5/53	53/5	9.4	95 @ 6000	98 @ 3800
131B2000	84 x 90	8.9:1	77–82	1995	41.8/36	5/53	53/5	9.9	Around 115 @ 5600	Around 128 @ 3600
131C4000	84 x 90	8.9:1	82–85	1995	41.8/36	5/53	53/5	9.9	Around 115 @ 5600	Around 115 @ 5600

Engine tuning notes:
All models have late pattern con-rods, but 1600 will not accept race bearings.
1600 has smaller (200mm) flywheel than 2*l*.
All models were fitted with late bellhousing and gearbox, 2-door Sport has Abarth remote gearshift.
1600 has cast crank, others are Tufftrided steel forgings.
All models have block-mounted distributor.
Rare 2*l* 16v Gp 4 models highly sought after.
Engine tuning potential: 1585 – 6/10; 8v 2*l* – 9/10.

FIAT 105/130 TC ENGINE DATA

SERIES	BORE/ STROKE (mm)	CR	YEARS	CAPACITY (cc)	VALVES (mm)	CAM TIMING (deg) IN	EX	LIFT (mm)	POWER (bhp DIN)	TORQUE (lbf ft)
138AR000 (105 TC)	84 x 71.5	9.3:1	83–88	1585	43.5/36	10/48	53/5	9.6	105 @ 6100	103 @ 4000
138AR2000 (130 TC)	84 x 90	9.45:1	83–88	1995	43.5/36	7/51	51/8	10	130 @ 5900	136 @ 3600

Engine tuning notes:
105 TC, Regata 100S, Prisma and Delta 1600 GT engines are essentially all the same and use cast crank.
130 TC has nitrided steel crank.
Transverse units can be converted to RWD by change of sump, oil pump and use of bearing in end of crank (input shaft of box may need to be shortened). All engines mounted vertically.
Engine tuning potential: 1585 – 7/10; 2*l* – 9/10.

LANCIA BETA ENGINE DATA (DONOR ENGINES AVAILABLE FROM COUPE, SPIDER, SALOON, HPE)

SERIES	BORE/ STROKE (mm)	CR	YEARS	CAPACITY (cc)	VALVES (mm)	CAM TIMING (deg) IN	EX	LIFT (mm)	POWER (bhp DIN)	TORQUE (lbf ft)
138AR000	84 x 71.5	9.3:1	83–88	1585	43.5/36	10/48	53/5	9.6	105 @ 6100	103 @ 4000
828B000	84 x 71.5	9.4:1	75–84	1585	41.8/36	13/45	49/9	9.4	100 @ 5800	103 @ 3000
828B1000	84 x 90	8.9:1	75–80	1995	41.8/36	13/45	49/9	9.4	115–119 @ 5500	134 @ 2800
828B4000 (*ie* version)	84 x 90	8.9:1	80–84	1995	41.8/36	13/45	49/9	9.4	122 @ 5500	135 @ 2800
828B7000 (Volumex version)	84 x 90	7.5:1	82–84	1995	43.5/36	13/39	37/3	9.2 IN 8.6 EX	135 @ 5500	152 @ 3000

Engine tuning data:
1585 cast crank, others steel. Volumex has sodium-cooled exhaust valves.
134AS Lancia Monte Carlo engine similar to 828B1000, mid-engined layout, others FWD.
Gearboxes all interchangeable, ratios common to all models, final-drive ratios vary. Volumex also used on limited number of 2*l* Spiders (Pininfarina Spider Europa). All engines 20° rear tilt.
Engine tuning potential: 1585 – 6/10; 2*l* (carb) – 9/10; 2*l* (*ie*) – 7/10; Volumex – 10/10.

LANCIA DELTA ENGINE DATA (SEE ALSO 105TC)

SERIES	BORE/ STROKE (mm)	CR	YEARS	CAPACITY (cc)	VALVES (mm)	CAM TIMING (deg) IN	EX	LIFT (mm)	BOOST (Bar-gauge)	POWER (bhp DIN)	TORQUE (lbf ft)
831A7000 Delta HF Turbo (carb)	84 x 71.5	8:1	83–85	1585	41.8/36	0/40	40/0	9.1 IN 8.6 EX	0.52	130 @ 5600	147 @ 3500
831B3000 HF (*ie*)	84 x 71.5	8:1	86–87	1585	43.5/36	0/40	40/0	9.1 IN 8.6 EX	0.8	140 @ 5500	147 @ 3500
831B5000 (Delta 4WD)	84 x 90	8:1	86–87	1995	43.5/36	8/42	41/1	9.1 IN 8.6 EX	0.9	165 @ 5250	196 @ 2500
831C5000 Delta HF Integrale (8v 4WD)	84 x 90	8:1	88–89	1995	43.5/36	8/42	42/1	9.1 IN 8.6 EX	1	185 @ 5300	234 @ 3500
831D5000 Integrale (16v 4WD)	84 x 90	8:1	89–91	1995	34.5/28.5	8/35	30/0	8.6 IN 7.5 EX	1	200 @ 5500	234 @ 3000

Engine tuning notes:
1585 models cast crank, 2*l* forged steel. Port layout changed '86 on.
All models have sodium-cooled exhaust valves. All models except 831A7000 have unleaded ex valve seats.
831A7000 uses Microplex ignition, all others have 'fully mapped' ignition/injection.
831A7000 is vertical, others 20° forward tilt.
2*l* models have larger clutch, *ie* 230mm friction face dia, than early 2*l* normally aspirated models; 1585 types use same dia clutch as 2*l* 131.
Engine tuning potential: 10/10.

'I wonder,' he said to himself presently, 'I wonder if this sort of car starts easily?'

Next moment, hardly knowing how it came about, he found he had hold of the handle and was turning it. As the familiar sound broke forth, the old passion seized on Toad and completely mastered him, body and soul. As if in a dream he found himself, somehow, seated in the driver's seat; as if in a dream, he pulled the lever and swung the car round the yard and out through the archway; and, as if in a dream, all sense of right and wrong, all fear of obvious consequences, seemed temporarily suspended. He increased his pace, and as the car devoured the street and leapt forth on the high road through the open country, he was conscious that he was Toad once more, Toad at his best and highest, Toad the terror, the traffic queller, the Lord of the lone trail, before whom all must give way or be smitten into nothingness and everlasting night. He chanted as he flew, and the car responded with sonorous drone; the miles were eaten up under him as he sped he knew not whither, fulfilling his instincts, living his hour, reckless of what might come to him.

From THE WIND IN THE WILLOWS
by Kenneth Grahame

CHAPTER 2

TUNING THEORY

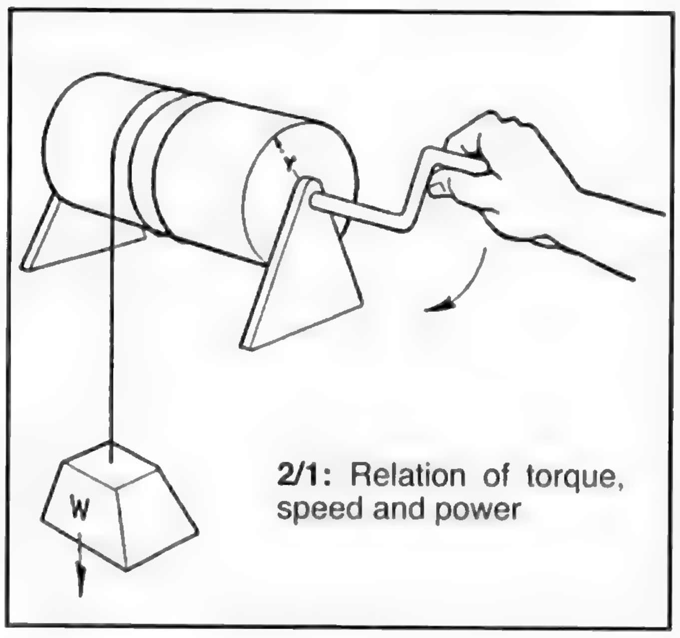
An engine is often referred to as 'powerful', but what does this mean? In real terms, it usually implies that it gives good acceleration and can sustain a high top speed. Is this high power, or torque? In fact, it is high torque that leads to good performance, both under the transient, accelerative mode, and at steady-state high speed, but the influence of torque is sometimes neglected in discussion. At this early stage, it would be appropriate to state that the 'science' of tuning is to raise the torque output throughout the broadest possible engine speed range. Power is merely a function of torque/rpm, and this relationship will be discussed in detail later. Some definitions at this point are important.

Work
Work is done when a force moves its point of application through a certain distance. No matter how great the force exerted, if there is no movement, there is no work done.

Power
Power is the rate at which work is done. To lift a given weight through a given distance implies a fixed amount of work. To lift it the same distance, but in a shorter time, implies the same work in less time. So more power is used, but for a shorter time to do the same work.

Torque
Torque is the 'twisting effect' in a shaft. When tightening a bolt, the shear effect caused by the twisting depends on the length of the wrench used and the force applied. So Torque = Force applied × Radius of application.

Relationship of torque, speed and power (2/1)
Torque = Force (W) × radius (r), or T = Wr.
One revolution of the windlass raises the weight a distance of 2π
ie Work done = Force × Distance moved $= W \times 2\pi r = 2\pi T$
If the windlass is turned through n



2/1: Relation of torque, speed and power

revolutions in unit time
work in unit time $= 2\pi T \times n$
ie Power $= 2\pi nT$

Units
Work (SI)
The standard units are:
mass – kilogram – kg
time – second – s (sec)
length – metre – m
force – Newton – N
where (from the equation Force = mass × acceleration):
$1N = 1kg \times 1m/s^2$
The work unit is 1 Newton operating through 1 metre, giving:
1 Newton metre – 1Nm, which is equivalent to exactly 1 Joule – 1J. Since the Joule is very small, the kiloJoule kJ is normally used.

Work (imperial)
The standard units are:
mass – pound – lb
time – as SI
length – foot – ft
force – pound force – lbf
where $1lbf = 1lb \times 32.18ft/s^2$ ($32.18ft/s^2$ is the acceleration due to gravity).
The work unit is 1 pound force operating through 1 foot, giving 1 foot pound force – lbf ft (or ft lbf).

Power (SI)
The power unit is 1 Joule per second, which is called 1 Watt – 1W. Again the kiloWatt kW is more usual. As explained earlier, the product of power and time is work, thus: 1kW for 1 second = 1kJ and 1kW for 1 hour = 1kW hr (3600kJ). (This is the unit by which electricity is purchased.)

Power (imperial)
The power unit is 550 pound force per second, called 1 horsepower – hp. Since again, the product of power and time is work, 1 horsepower for 1 hour = 1hp hr = 3600hp second $= 1.98 \times 10^6$lbf ft. Horsepower measured on a dynomometer is referred to as brake horsepower, or bhp.

Useful conversions

Torque	1lbf ft = 1.36Nm
	1Nm = 0.74lbf ft
Power	1bhp = 0.75kW
	1kW = 1.34bhp

Typical example (British units):
If a 2l TC develops 198bhp at 7500rpm, what is the torque output?

$$\text{Power} = 2\pi nT, \text{ or, rearranging } T = \frac{\text{Power}}{2\pi n} \quad (\pi = 3.142)$$

where: T = Torque, n = engine speed (rev/min)

hence: $$T = 198 \times \frac{1}{2\pi} \times \frac{1}{7500} = 4.20 \times 10^{-3} \text{ bhp min}$$

or more conventionally:

$$T = 4.2 \times 10^{-3} \text{ bhp min} \times 550 \times 60 = 138.6 \text{ lbf ft}$$

A useful quick form of this calculation to remember is:

$$\text{Power (bhp)} = \frac{T \text{ (lbf ft)} \times n \text{ (rpm)}}{5250}$$

Typical torque (SI units):
If a 2*l* TC develops 130kW at 7400rpm,

$$\text{Power} = 2\pi n T,\ T = \frac{\text{Power}}{2\pi n}$$

where units are: Power – Watts
Speed – rev/sec
T – Nm

$$\text{Torque} = \frac{130 \times 10^3 \times 60}{2\pi \times 7400} = 167.7\ \text{Nm}$$

Torque or power?
From the equation:

$$P = 2\pi n T$$

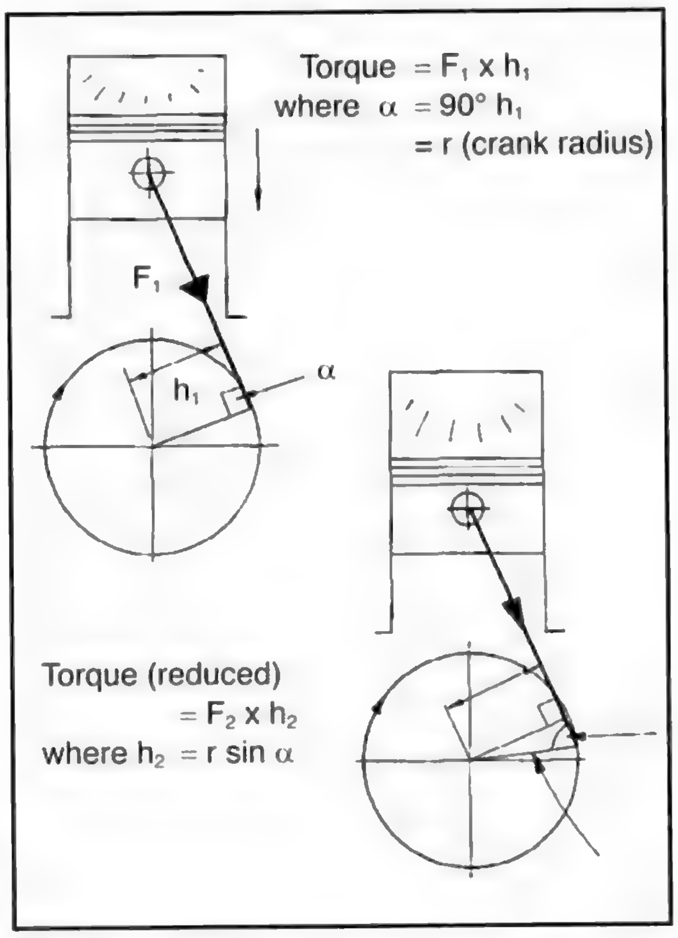
it is obvious that, if torque was constant (*ie* did not vary with rpm), the power output would increase linearly with engine speed. In practice, for reasons which will become clear in due course, the torque output varies with rpm, though to a lesser extent with pressure-charged engines, and the exact power output at any speed will depend on this. Very high torque at low speed may give the same power output as very low torque at very high speed, and similarly modest torque at low speed will give a poor power figure, but this does not imply that acceleration will be poor. Acceleration is produced by torque. To *sustain* a high top speed against the effects of inertia, gradient type, rolling resistance and aerodynamic resistance requires a high rate of work, *ie* high power.

Influences on power/torque output
Torque variation
A brief analysis of the working cycle of the engine shows that fuel/air mixture (petrol/air) is inhaled and burned to produce pressure in the cylinder. This gas pressure acts on the piston to force the assembly downwards in turn giving rise to rotation of the crankshaft. From the equations:

$$\text{Force} = \text{mass} \times \text{acceleration}$$

$$\text{and Gas pressure} = \frac{\text{gas force}}{\text{piston area}}$$

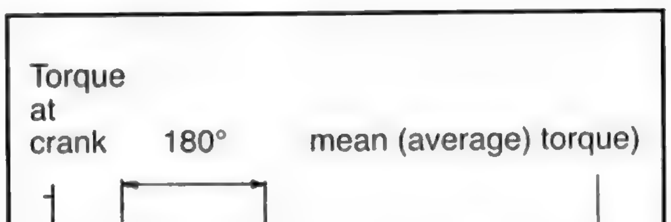
the greater the gas force *(see Chapter 9 – Pistons and Rings for analysis of gas force)* the greater the acceleration of the piston and vehicle as a whole. This process, of course, does not take place entirely unopposed, because to accelerate any body from rest requires a certain proportion of the applied force to be used up in overcoming inertia – the resistance of a mass to motion – and the effects of friction. The net force acting on the piston is the product of gas force minus inertia force and frictional force. What is left (after these deductions) is available to perform 'useful' work.



2/2: Variation of torque arm (h) with crank rotation.

The rotation of the crankshaft being generated by the reciprocating motion of the con-rod, the torque at the crankshaft is determined by the product of this net force acting along the rod axis and the radius at which it is applied **(2/2)**.

Added to the effect of the variation of the length of the torque arm 'h' the gas force 'F' varies. Firstly, the engine inhales a widely varying amount of charge according to its design characteristics and throttle position/speed. Added to this, the net gas force varies with piston position – as the cylinder volume increases the pressure drops, and because of the sinusoidal motion of the piston (it performs a simple harmonic motion) its acceleration varies – even at constant engine speed – so the inertia force (opposing the gas force) changes too. The effect of this is to produce a fluctuating, or cyclical, torque output **(2/3)**.



2/3: Four-cylinder cyclical torque variation.

Since each cylinder of the four-stroke engine fires only once every two crank revolutions, the four-cylinder engine requires 720° of crank rotation for all cylinders to complete their full working cycle. The mean torque output is measured at the flywheel on a brake dynamometer, and it is worth noting at this stage that the flywheel itself has an influence on the mean torque since it can store energy between firing strokes (180° interval on the four-cylinder engine).

Conversion of fuel into power
The net force on the pistons during the firing strokes determines the turning torque transmitted to the flywheel. Shortly after ignition, the cylinder pressure rises to a maximum – peak cylinder pressure – then drops as the piston moves down the bore until, when the exhaust valve opens, the cylinder pressure drops to the point where no further useful work is performed. Peak cylinder pressure depends on a number of interrelated factors, and its measurement is carried out by means of pressure indicator systems which map the pressure-volume relationship in the cylinder. Whilst interesting in itself, the main value of this measurement is to derive a value for the indicated mean effective pressure (imep), which can be used as a basis for calculation and analysis of the engine torque output. There is a variation between the theoretical, *ie* indicated mean effective pressure and the brake mean effective pressure (bmep), which generates usable torque at the flywheel, due to mechanical losses – friction/inertia. In other words, the power output at the flywheel is governed by the following relationship:

Indicated power = brake power + mechanical losses (where the mechanical losses are often referred to as 'friction horsepower', Fhp).

η_{mech}, mechanical efficiency (%)

$$= \frac{\text{brake power}}{\text{indicated power}} \times 100$$

$$= \frac{\text{bp}}{\text{bp+Fhp}} \times 100$$

Some dynomometers are capable of measuring the 'friction power' (Fhp), and thus the mechanical efficiency (ηmech) of the engine can be calculated; an analysis of ηmech is a useful method of determining just how much usable power is being lost.

Bmep and imep are related to the power output of the engine according to the following equations *(see next page)*:

$$\textbf{indicated power, ip} = \frac{p\,L\,A\,n\,e}{2\pi}\ (\text{kW})$$

$$\text{and } \textbf{brake power, bp} = \eta_{mech} \times \frac{p\,L\,A\,n\,e}{2\pi} (\text{kW})$$

where:
p = indicated mean effective pressure (kN/m^2)
L = stroke (m)
A = piston area (m^2)
n = engine speed (radians/sec)
(100 rev/sec = 100 × 2π rad/sec)
e = number of power strokes per revolution
(for the 4-cyl TC e = 2)

$$\text{since } \eta_{mech} = \frac{bp}{ip}, \text{ and also } \eta_{mech} = \frac{bmep}{imep}$$

the above equation can be written as:

$$\textbf{brake power} = \frac{bmep \times L\,A\,n\,e}{2\pi}\ \text{kW}$$

Thus, if the brake power, L, A, n, e are known, the bmep can be calculated. The peak value will correspond with the speed at which maximum torque is produced, and examination of bmep is useful in the search for more torque (throughout the range) in the sense that the higher the bmep, the better the torque.

The indicated pressure on the piston depends on a number of closely inter-related factors. Firstly, high volumetric efficiency, *ie* good filling of the cylinder with air/fuel mixture, is required – the greater the mass of charge (in the correct ratio of air-fuel) the more energy is available. Secondly, the efficiency of the combustion cycle must be such that this charge is effectively burned to produce as high a pressure as possible. This efficiency depends partly on the design of the combustion chamber and means of ignition initiation (in the right place in the cycle) and partly on the compression ratio (CR) – the relationship between the volume in the cylinder with the piston at TDC and the swept volume of the cylinder. Not only does the CR have a major bearing on the efficiency with which the engine converts fuel into power (thermal efficiency – or η_{th}), it also affects the peak pressure achievable during the firing stroke in that the smaller the volume into which the charge is compressed prior to ignition, the greater the peak pressure that will result. The CR, computed from the dimensions of the clearance and swept volumes, will represent the 'static' compression ratio – on an efficient engine the 'effective' CR may be much higher – and on pressure-charged engines higher still, as the charge enters at a pressure above atmospheric. In this sense, the ability of the engine to inhale fresh charge (volumetric efficiency – or η_{vol}) has an important influence on CR and cylinder pressure as a whole.

Typical bmep
2*l* GCT NHRA Fiat:

$$\text{brake power} = \frac{bmep \times L\,A\,n\,e}{2\pi}\ \text{kW}$$

at peak torque, 5000 rev/min,

$$108.4 = bmep \times \frac{0.09 \times 0.0057 \times 83.3 \times 2\pi \times 2}{2\pi}$$

$$= bmep \times 0.09 \times 0.0057 \times 83.3 \times 2$$

$$\text{hence bmep} = \frac{108.4}{0.085} = 1268.3\ \text{kN/m}^2\ (184\ \text{lbf/in}^2)$$

(*Note:* To convert kN/m^2 to lbf/in^2, convert to N/m^2 (multiply kN/m^2 by 1000) and divide by 6895.)

This may not actually represent the ultimate bmep from an 8v TC, but certainly it is a respectable figure by any standards, which can probably be enhanced by closer attention to volumetric efficiency, CR, combustion efficiency and mechanical losses. A normally aspirated 16v engine of similar dimensions might reasonably give (developing 178lbf ft torque at 5500rpm) 212lbf/in^2, principally, GCT believe, due to the high volumetric efficiency of the valve layout, and the superior combustion efficiency of the (low-volume) 16v head.

INERTIA EFFECTS

During the power stroke, the reciprocating and rotating masses of the engine are accelerated by the force acting on the piston. As the power stroke of each cylinder comes to an end, the momentum generated in these components carries the piston beyond BDC and upwards on the exhaust stroke. On the four-cylinder engine the firing stroke of the next cylinder is just commencing at this point – the more cylinders an engine has, the lower the requirement for this momentum effect to keep the engine running smoothly.

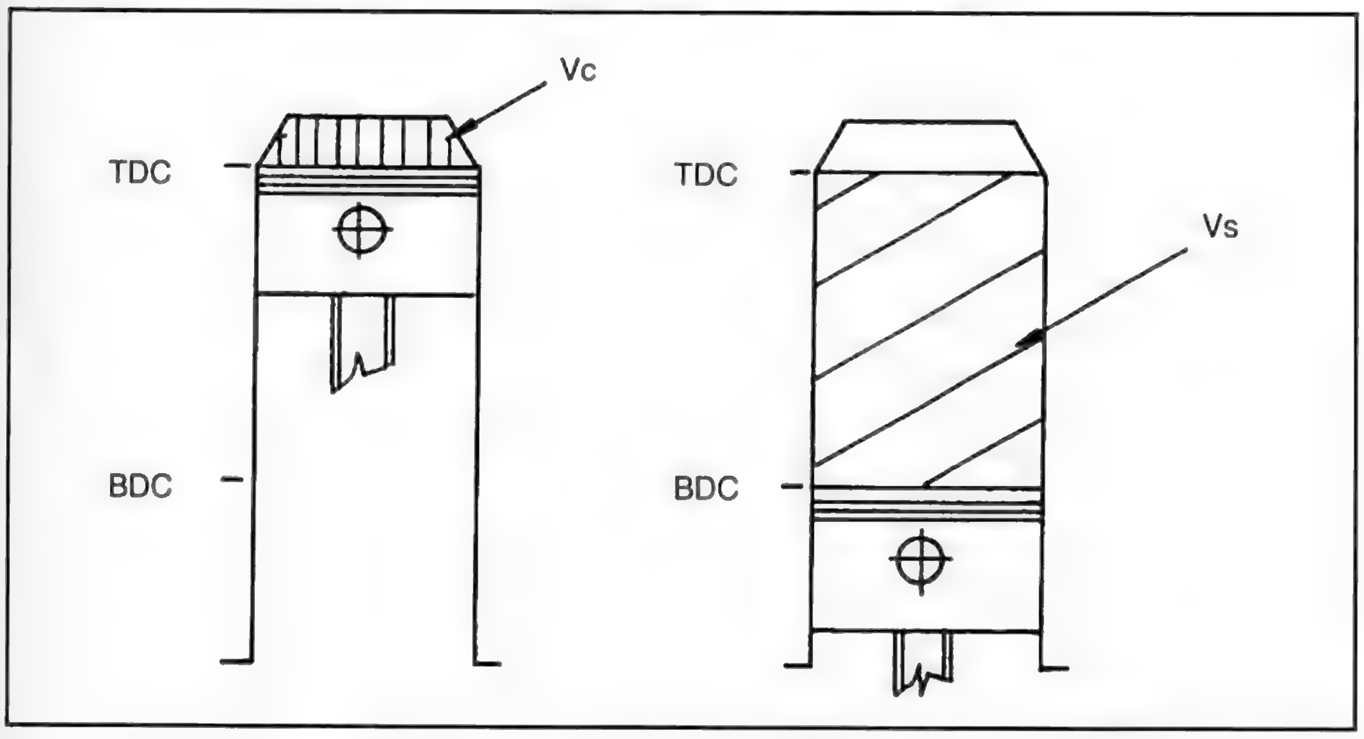
However, no mechanical system is 100% efficient – in other words, the energy returned to the system will always be less than the input amount. If this were not true and there were no losses, a flywheel, once set in motion, would spin forever! It is therefore important that the lightness of these assemblies is carefully matched to the requirements of the engine – GP engines use virtually no flywheel and a very small diameter clutch because their multi-plane (v) crankshaft layout allows the multiple pistons to keep the engine rotating smoothly without the need for external energy storage. Since work is lost, *per se*, in driving these components, albeit that some is returned to the system, overly heavy reciprocating/rotating masses will lead to a torque loss at the flywheel, even at steady speed, against an engine with lighter components.

In addition, much of the time the crankshaft of a competition engine is accelerating in order to achieve a high power output, and the lighter the components, the faster this will take place. It is also worth noting that the heavier the components, the greater will be the frictional losses inside the engine.

Compression ratio (CR) is defined as:

$$CR = \frac{\text{swept volume (Vs) + clearance volume (Vc)}}{\text{Vc}}$$

where Vc is the volume in the cylinder above the piston at TDC and Vs is the total volume swept by the piston between TDC and BDC **(2/4)**.



2/4: Compression ratio (CR).

Volumetric efficiency (η_{vol}) is defined as:

$$\eta_{vol} = \frac{\text{actual mass of air inhaled during intake stroke}}{\text{mass of air to fill swept volume at ambient temp/pressure}}$$

Clearly, the energy of the fuel is an important feature. The calorific value of the fuel determines its energy content in terms, for example, of the kW of power that can be obtained (theoretically!) from a given quantity. The amount of fuel needed by an engine to develop horsepower is defined by its thermal efficiency. As with mep, the definition of η_{th} is broken into two parts:

- Indicated thermal efficiency

$$\text{ind } \eta_{th} = \frac{\text{indicated power output}}{\text{rate of energy supply of fuel}}$$

- Brake thermal efficiency

$$\text{brake } \eta_{th} = \frac{\text{net power output (brakepower)}}{\text{rate of energy supply of fuel}}$$

where the indicated and brake values of power are a function of imep and bmep respectively. It is an unfortunate fact of thermodynamics that the brake thermal efficiency of petrol engines is quite poor – usually around 26% at best for a normally aspirated engine, although turbocharged engines, utilizing spent energy from the exhaust to increase the charge density, have a fractionally higher rating except 'off boost', when the thermal efficiency is reduced by the low CR. η_{th} varies marginally according to engine speed and load and is primarily influenced by the compression ratio. Brake thermal efficiency is essentially a measure of the mechanical losses (friction/inertia) in the engine, which reduce the indicated power output.

The relationship between indicated η_{th} and brake η_{th} can also define the mechanical efficiency of the engine, *ie*

$$\text{mechanical efficiency } (\eta_{mech}) = \frac{\text{brake } \eta_{th}}{\text{ind } \eta_{th}}$$

The mechanical efficiency of a well-built TC engine can exceed 90% at low speed, but even *indicated* thermal efficiency is always vastly lower than100% because of energy losses to coolant, radiation and exhaust (some 70%) and energy lost in the pumping cycles of induction and exhaust **(2/6)**.

The efficiency of these latter phases can be enhanced, but there is little that can be done about the heat loss although results from thermal coatings on combustion chambers, pistons and exhaust ports appear to yield measurable power increases. Added to this, the conversion of fuel into energy (in the case of petrol) is far from perfect because of the phenomenon of dissociation: some of the constituents of the combustion process, CO_2, O_2 and H_2, are converted into their component parts (C, CO, O, H, etc) at high temperature: this chemical process absorbs energy.

SWIRL-SQUISH

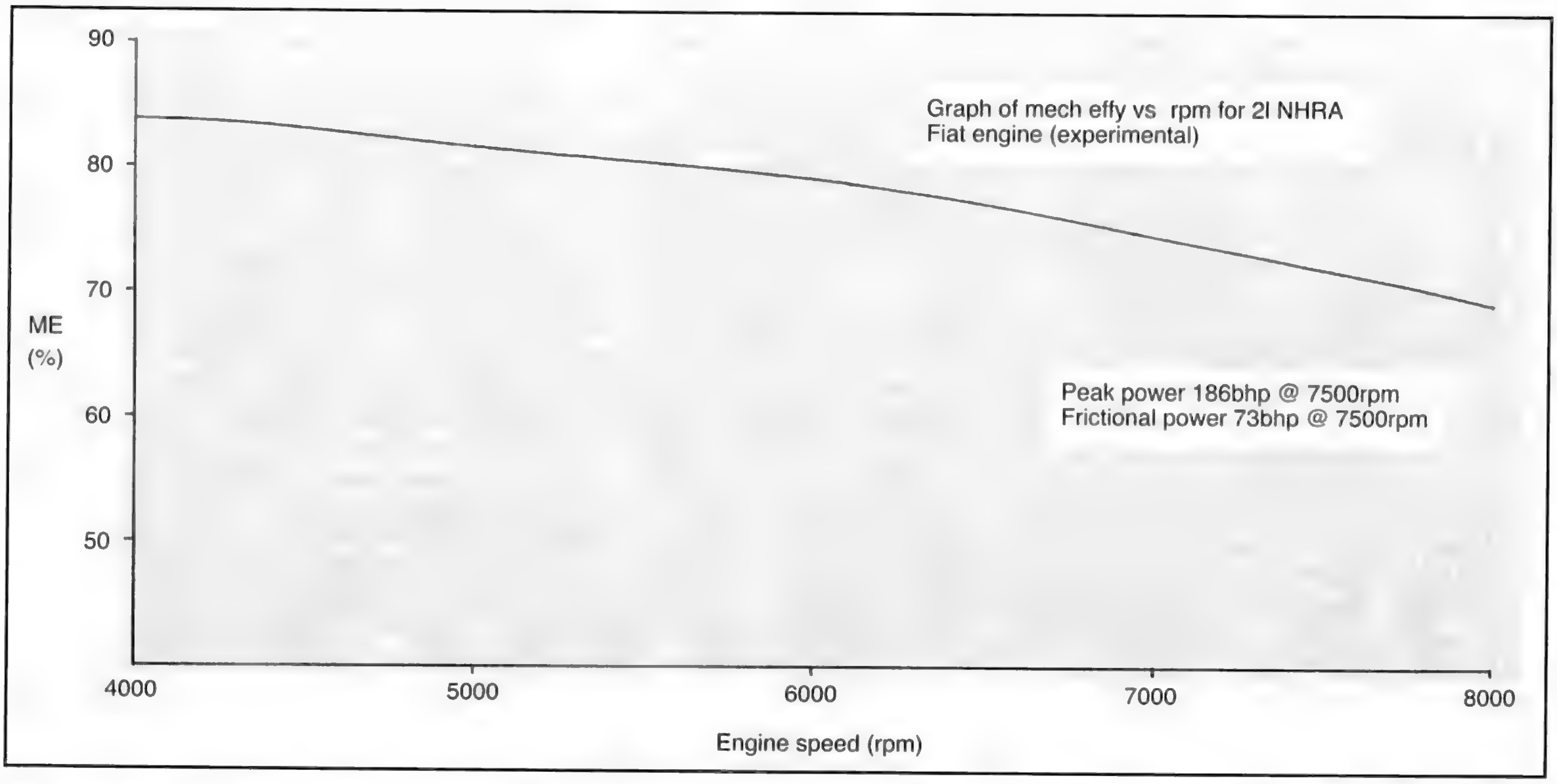
The creation of a controlled amount of turbulence in the combustion chamber prior to ignition improves the burn rate of the fuel – leading to higher cylinder pressure just when it is needed. This motion of the fuel/air mixture is known as swirl turbulence, and in the TC it is primarily induced by means of 'squish bands' *(see diagram)* created by the close proximity of the piston crown to the combustion chamber **(2/5)**. (Other engine designers arrange the port layouts so that the incoming mixture naturally adopts a swirling motion as it enters the chamber.) Swirl turbulence can also be induced by 'swirl polishing' the inlet valves – a mildly coarse spiral ground finish on the back of the valves.

Swirl has the additional benefit that it reduces the tendency to detonation, and examination of the 8v turbo engines will reveal quite large squish bands.

Production normally aspirated TC squish bands vary from 2mm to about 1.65mm. GCT have used as little as 1mm, the only drawback being that the confined space (dead volume) tends to trap unburnt hydrocarbons. Whilst insignificant from a tuning perspective, it is worth mentioning that the motor industry is increasingly concerned with unburnt HC levels, both from a pollution and a thermal efficiency point of view.

Swirl becomes less important at very high engine speeds, and on big-valve TCs removal of material from the squish band to deshroud the valves should take priority.

2/5: Squish band.



2/6: Mechanical efficiency versus rpm.

Conversion of the fuel into useful power also requires as much of the charge to be consumed during the combustion process as possible, minimizing the discharge of unburned hydrocarbons during the exhaust stroke. This conversion relies heavily on optimization of the combustion chamber layout and fuel system.

Volumetric efficiency determines, in the main, the torque and power output from the engine in that the more charge is induced (in the correct air-fuel ratio) the more cylinder pressure will result. It follows that, since thermal efficiency is more or less fixed at a low level, although more torque/power can be obtained from an engine by special tuning, there is no escaping the fact that obtaining more power will lead to an increase in fuel consumption in terms of the fuel used/power unit power increase.

Merely introducing more fuel into the cylinder will not, *per se*, lead to more power since the ratio of air to fuel must be maintained at an optimum level. Chemically correct, in theory, is 14.7 parts by mass air to fuel, but because of dissociation it is necessary, for best power, to use a mixture nearer 14.2–14.5:1. (When the engine is cold, only the lighter fractions of the petrol will vaporize (and burn) in the cylinder; on cold starts anything from 3:1–8:1 may be needed, depending on ambient air temperature.)

TCs – typical thermal efficiency

2.1l Fiat, GC 3A cams, 11:1 CR, 45 DCOE (38 choke), 43½/36 valves.
Max power 135kW @ 7000rpm. Max torque 200Nm @ 5500rpm. Power at max torque 115.3kW. (Full-throttle figures.)

Brake specific fuel consumption

at 5500rpm	316 gm/kW hr
at 7000rpm	320 gm/kW hr
Fuel calorific value	43,960 kJ/kg

η_{th} At max torque

Mass rate of fuel consumption

$$= \frac{316\text{gm/kWhr} \times 115.3\text{kW}}{1000}$$

$$= 36.43\text{kg/hr (fuel consumption at max torque)}$$

Energy input from fuel

$$= 43960\text{kJ/kg} \times 36.43\text{kg/hr} \times \frac{1}{3600}\ \text{kJ/s}$$

$$= 444.85\text{kJ/s} = 444.85\text{kW}$$

Thermal efficiency

$$= \frac{\text{useful work output}}{\text{energy supplied in fuel}} = \frac{115.3}{444.85} = 26\%$$

η_{th} At max power

Mass rate of fuel consumption

$$= \frac{320 \times 135}{1000} = 43.2\text{kg/hr}$$

Energy input from fuel

$$= 43960 \times 43.2 \times \frac{1}{3600} = 527.52\text{kW}$$

Thermal efficiency

$$= \frac{135}{527.52} = 26\%$$

TC SPECIAL TUNING

Raising the volumetric efficiency

The theoretical maximum mass of air that the engine can inhale is normally assumed to be equivalent to that required to fill the swept volume. In practice, volumetric efficiencies of over 100% can be achieved on well-tuned normally aspirated TCs and, of course, the boosted (turbocharged and supercharged) versions can achieve 200%-plus, and this improvement translates directly into more torque.

Conversely, when the engine is operating outside its optimum range the value of η_{vol} will suffer because the engine cannot inhale enough air, and the torque will fall off. Good volumetric efficiency leads to high cylinder pressure and thus high exhaust energy, at the same time 'scavenging' the cylinder of exhaust residuals. All these factors lead to better torque.

Increased power calls for more cycles per second, so it is vital that the volumetric efficiency is not only good at low speed, but also at high speed. To achieve this requires an appreciation of the fact that complex pressure wave effects are present in the inlet tract, and that it is vital to sustain as much air momentum as possible at all times – even when the inlet valve is closed (over half its cycle, even with race cams).

If exhaust residuals remain in the cylinder they will both contaminate the fresh charge and reduce the charge mass by restricting its entry into the cylinder. If the air/fuel ratio is excessively rich, the same result will occur because the excess fuel will displace air. The intake air must be of a temperature that allows the fuel to vaporize in the inlet tract. With fuel injection the fuel is more finely atomized than with carburettors, and thus cold intake air can be used – raising the charge density and η_{vol}. The use of intake air at the right sort of temperature is quite crucial to power output – the power loss from excessive inlet temperature can be as much as 2bhp per °C.

Much care, during special tuning, should be taken to ensure as high a value of η_{vol} as possible. In summary, the following factors should be taken into account:

1. High-efficiency air filter to minimize pressure drop across element.
2. Use of cold air ducting to reduce air charge temperature and raise density.
3. Rampipes should be carefully designed to reduce turbulence (and resulting flow loss) at inlet tract entry.
4. Inlet tract and carb/fuel injection system (including choke, throttle plate, etc) should be shaped for maximum ram effect (to sustain air momentum) and minimum turbulence.
5. The valve/throat area must be shaped to ensure minimum interruption of the flow into the cylinder.
6. The combustion chamber and piston must not interfere with the inlet/exhaust flow cycles.
7. Camshafts with appropriate lift and duration must be used and their characteristics must match the inlet/exhaust flow characteristics of the cylinder head to produce torque in the right place.
8. Separating the inlet tracts (*eg* one carb choke per cylinder) greatly improves the air momentum effect by reducing interference between cylinders.
9. Use of larger valves or multi-valve heads improves flow rates of inlet/exhaust.
10. The use of a properly designed exhaust manifold can greatly enhance cylinder scavenging.
11. A high CR contributes to high volumetric efficiency.
12. Use of a fuel-injection layout removes the requirement for a choke and raises the volumetric efficiency *per se* (by about 10%). Carbs can be modified (by knife-edging the throttle plate and careful internal blending). Slide-throttle injection (no butterfly – hence zero airflow restriction at full-throttle) raises η_{vol} still further.

In practice, measurement of engine volumetric efficiency can be carried out on a dynamometer; improvements to the TC layout and flowbench testing to raise the mass flowrate of air (cu ft/min) are dealt with extensively in later chapters.

PRE-IGNITION AND DETONATION

A discussion of these effects is valid at this stage since they have an important influence on compression ratio. Both these effects result from combustion problems and can have catastrophic results. They lead to a sudden rise in temperature and pressure in the cylinder.

Pre-ignition ('pinking')

Causes:

Hot-spots caused by carbon build-up on head or piston, overheating plugs, or heat build-up in sharp projections in the chamber.

Blown head gasket, causing flame in one cylinder to ignite mixture early in adjacent cylinders.

Symptoms:

Audible noise (not unlike the sound of a small hammer striking an anvil).

Power loss.

Possible smoking from exhaust.

Engine temperature rise, engine runs-on after switching off.

Effects:

Damage to pistons, head, cylinder head gasket.

Detonation

Causes:

Fuel octane too low, CR too high, leading to uncontrolled combustion.

Ignition over-advanced, and very high peak cylinder pressure.

Inlet air temperature too high.

Poor fuel distribution or over-lean mixture.

Poor combustion chamber layout.

Effective CR (race cams) too high (excessive cylinder pressure).

Symptoms:

Audible noise (as pre-ignition), sometimes hard to detect because of engine noise.

Power loss.

Possible smoking from exhaust.

Engine temperature rise (hot-spots may be created from this and lead to running-on, as with pre-ignition).

Effects:

Piston crown damage, broken ring lands, head damage.

Of the two effects, detonation is usually the more serious. The problem is especially acute with turbocharged engines. Late (post-Delta 1.6 *ie*) TCs are equipped with knock sensors to retard the ignition when the sensor picks up the tendency. Detonation can punch through pistons in seconds on turbocharged engines. On normally aspirated engines the high cylinder pressures and temperature cause the material of the head and/or piston to go 'pasty'. The shock waves from detonation can also shatter the piston rings and break the ring lands.

A poor combustion chamber design can create detonation by trapping 'end gas' which, if excessively overheated by heat transfer from the combustion chamber or piston, can be ignited by the shock wave from ignition, creating two flame fronts, which collide, leading to excessive cylinder pressure.

Raising the CR

Increasing the compression ratio does two things: it makes the engine more thermally efficient and it raises the cylinder pressure.

The use of competition camshafts can reduce the compression ratio when they are not working in their optimum power band, because the late closure of the inlet valve allows inlet air (which at low speed has a low momentum) to be forced back out of the inlet tract. (Ideally, competition camshafts should only be fitted to an engine of raised CR – if the CR is too low, there will be a 'hole' in the torque curve with possibly only a modest increase in peak power at higher rpm.) The practical limit used so far by GCT (with suitably mapped ignition) is 13.7:1 static; this is primarily determined by the resistance of the fuel to detonation, therefore fuel octane rating must be established prior to adoption of a higher-than-standard CR. However, with a high CR, there is a much higher risk of detonation from the effects mentioned earlier, even when the octane rating of the fuel is adequate, so it is vital to ensure that the design of the new engine is carefully considered, especially in terms of its mechanical strength (particularly pistons – a high cylinder pressure puts greater stress on the ring lands), the fuel system/mixture distribution, and the ignition timing at various speeds/loads. The TC is inherently very resistant to detonation, partly since it employs an alloy head, and partly due to the design of the combustion chamber – with broad squish bands to induce horizontal swirl, which inhibits the tendency.

Current UK fuel octane ratings (Research Octane Number – RON) are:
Unleaded – 96 octane
Four-star – 97 octane
Super unleaded – 98 octane
Five-star – 99 octane (currently only available at certain race circuits)
Avgas – 100 octane or higher (used successfully by GCT despite its lower volatility)

GCT-recommended maximum CRs to suit various engine set-ups and fuel are as follows:

Normally aspirated

96 octane –	9:1 CR max, standard cams only (twin carburettors can be fitted)
97 octane –	9.6:1 CR with up to St II cams (single or twin carbs)
	10:1 CR with standard cams (single or twin carbs)
98 octane –	11.5:1 CR with any cams (twin carbs only) (Above 11.5:1 mapped ignition preferred)

For fuel-injected engines, the CR can go at least 0.5 higher.
Single-carb use with high CR requires careful attention to jetting to avoid knock.

Turbocharged

96 octane –	8:1 CR up to 12lbf/in^2 boost
97 octane –	8:1 CR up to 15lbf/in^2 boost
98 octane –	8:1 CR up to 18lbf/in^2 boost
–	7.8 CR up to 22lbf/in^2 boost
–	7.6 CR up to 25lbf/in^2 boost

The above figures are derived from fuel-injected units; for carburetted units, lower the static CR by 0.2 for a given boost. A knock sensor is vital on all turbo engines. *Note:* Octane enhancers are available quite widely to raise the octane value of low-grade fuel.

Compression ratio for pressure-charged engines

This is effectively the opposite of normally aspirated engines, where the CR is lowered appropriately to allow for the level of boost. When the boost is raised above the manufacturer's specification, either the static CR must be lowered or higher-octane fuel used. The lower CR is to compensate for the much higher cylinder pressure generated by the boost, in order to help prevent detonation.

Raising engine speed

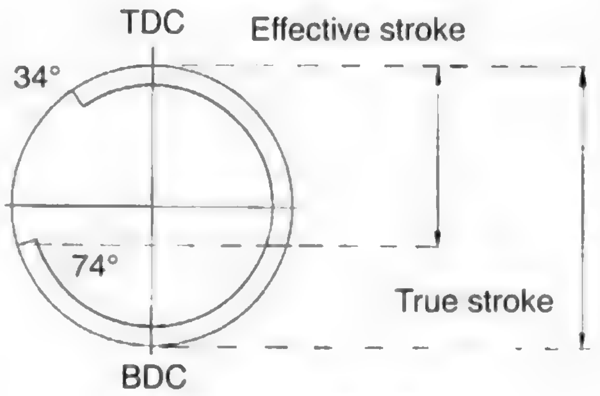
To obtain high power, a good high-speed torque characteristic is needed as mentioned previously: the engine must be made to produce this in the right place to match the gearing of the vehicle and the needs of the particular competition event. For example, a 2*l* circuit race car (in modified trim) could tolerate a power band from 5000–8500rpm (with 'nothing' below), but for hillclimbing it might be necessary to lower the power band to, say, 4000–7500rpm so that low gear/low road speed can be used.

COMPRESSION RATIO

The formula outlined earlier neglects the effect of inlet valve closure timing. A yardstick commonly used in racing for determining the optimum CR for a normally aspirated unit is to utilize a swept or clearance volume which will give a compression ratio of 9.5:1 based on the 'effective' stroke of the engine, measured from the point of inlet valve closure (2/7):

Example: With an effective stroke of 80mm, the swept volume of the 2*l* is 443.4cc and the Vc to give a CR of 9.5:1 must be calculated:

$$ie\ 9.5 = 1 + \frac{443.4}{Vc}\ ie\ Vc = 52.2cc$$



2/7: Inlet cam cycle.

Substituting this into the formula for CR leads to a calculation of the static CR required on the actual engine, *ie:*

$$CR = 1 + \frac{500}{52.2} = 10.6:1$$

LEADED OR UNLEADED FUEL

Petrol manufacturers in many countries (USA were the first) have started producing unleaded fuel because of the toxic danger of the lead content in the exhaust. Indeed, in the UK the lead content of UK 4-star (leaded) has been progressively reduced. Lead in the form of tetra-ethyl lead or tetra-methyl added to petroleum raises its octane rating and hence the resistance to detonation. This lead has the secondary benefit of lubricating the exhaust valve/seat, which rotates in service, thus ensuring that the contact face stays in good condition.

An engine which is run on unleaded fuel requires the use of alloy steel exhaust valve seat inserts to prevent the distortion of the seat inserts which would otherwise take place. Ideally, bronze or comparable exhaust valve guides should also be used because of their phenomenal capacity to disperse heat from the valve. Stellite coating of the exhaust valve contact faces and use of sodium-cooled exhaust valves are also desirable, but not essential ways of improving the longevity of the exhaust seat.

Fuel manufacturers now use alternatives to lead to raise the octane rating of their petrol, and in the UK both low (96 octane) and high (98 octane) fuel is available as well as the traditional 4-star (leaded) – now 97 octane compared with 98 in the early 80s.

Engines fitted with catalytic converters (or a Lambda-Sond oxygen sensor) must be run on unleaded fuel or damage will result.

Mostly, alterations to the torque/speed relationship come from the camshaft configuration, and since there is a finite limit to the torque that a normally aspirated engine can be made to produce, which falls off inevitably either side of the peak figure, the only way very high peak power can be produced is to combine as much available torque as possible with the highest possible rpm. Competition cams inevitably 'bolt-on' at the top, and 'take-away' at the bottom! Turbos are not confined to this constraint because pressure-charging flattens out the torque curve, potentially (with the right turbocharger and engine match) giving substantially more of it across a broader spread of rpm than its n/a counterpart.

Stress

This must be taken into account at an early stage. Some of the factors concerned are:

A Accelerative forces on	B Gas pressure load on	C Thermal stress on
piston crown pin boss rod small end/big end rod shank rod bolts bearings crank journals (mainly cast 1585) flywheel and bolts valve springs cam profiles	piston crown ring lands bearings head bolts and gasket rod as (A) head gasket spark plugs	exhaust valves pistons cyl head and gasket seals valve springs (esp turbos) turbocharger/ exhaust manifold

(Throughout this book, GCT have attempted to recommend the necessary attention to ensure reliability in these areas.)

Loadings A and B give rise to cyclical stress, which in the case of fasteners (*eg* rod bolts) is overcome (within the predicted life cycle of the item) by pre-stressing (torque-up) which applies a pre-load to the fastener which is in excess of the fluctuating load induced by the accelerative/gas pressure loads.

It is true to say that of all the components in the engine, the hardest working is the oil, which has to suffer attack from all three – and cope with being pumped around the engine at high speed at the same time – all the while subject to a violent, corrosive atmosphere! We owe the oil companies a debt of gratitude for their work in this field.

Optimizing mechanical efficiency

This is essentially a question of keeping reciprocating and rotating weights to a minimum and ensuring that friction and inertia are also kept to a minimum. Lightening (where practical – lightening the 2*l* crankshaft would have to be carefully considered in view of the cost involved) is dealt with subsequently; frictional losses can be minimized by adhering to the recommended dimensions specified later. There is a fine line between a 'loose fit' (*eg* crank bearings), leading to impact damage, and the 'correct fit', which will give both long life and free-running. Assessments of the turning torques of the various components should be made during the build-up.

Designing to meet race regulations

Prior to embarking on modification of the engine it is vital to examine the competition regulations for the particular event. Normally, first-level regulations are published by the event organizers and can be obtained on request.

These are supplemented in the UK by the more broad-reaching regulations published in the *RAC Motorsports Association (RACMSA) Yearbook* (the '*Blue Book*'). This excellent publication contains not only important details of tuning regulations for various formulae, but 'race legal' requirements appertaining to construction of cars and engines (including rollcages, safety cut-off switches, fuel/lubricant/coolant hose layouts, fire extinguishers, etc) and a wealth of useful information about race conduct, licences and useful contacts.

Competition at International level, *eg* Gp A or Gp N, requires an examination of the *FIA Yearbook* (the '*Yellow Book*'). The FIA *(Federation Internationale de l'Automobile)* is the world governing body for motorsport. Failure to read the '*Yellow Book*' regulations can have serious implications as it contains 'catch all' rulings not referred to in the '*Blue Book*' (such as – Gp N – 'No metal may be removed from the cylinder head')!

Over the years, numerous Fiat/Lancia cars have been homologated for use in competition in Gp A and Gp N International-level events. The papers relating to permissible engine/car modifications for these rally events can be obtained from the RACMSA.

Stage tuning

GCT have traditionally tended to look at the revised specification of the engine as a whole, *ie* Fast Road, St II (with competition cams), St III, St IV, etc. Many companies will offer stage-tuned heads with no guarantee that they will 'light up' on the client's block/engine assembly. However, the following specifications will give an idea of the kind of things that can be achieved:

Dyno test of 2*l* 'Fast Road' engine

This was the first dyno test by GCT to establish the torque output from a mildly tuned 2 *l* Fiat fitted with 45 DCOE, utilizing standard camshafts.

Full spec:
132 1800 pistons (9.6:1 CR) 84.4mm bore
Head fully ported, blueprinted seats, 42/36 valves (unmodified inlets)
GC sidedraught manifold, 45s, 36mm chokes
Cams timed at standard setting
Lightened flywheel, balanced with crank
Exhaust 4-1, 36" primary lengths

Output:

Speed (rpm)	Torque (corrected) (lbf ft)	Power (corrected) (bhp)
3750	142.8*	102
4000	140.7	107.2
4500	140.7	120.6
5000	140.7	134
5500	138.7	143.3
6000	134.5	153.7
6250	130.4	155.2*
6500	124.2	153.7

* max outputs

Jetting:
145 main
F16 emulsion tube
155 air corrector
55F8 idle jet
45 pump jet
Flexibility test:
Engine pulled full-throttle from 1800rpm.

This test was conducted on a manual-control dyno. In the light of subsequent tests, the peak torque figures can be regarded as being accurate to within 3% and the test demonstrates the

extraordinary ability of this engine to 'hold its torque'. Torque drops off quite rapidly over 6500rpm, but the torque characteristic between 3750 and 6250rpm makes engines of this type ideal for 'fast-road' use.

GCT OUTLINE SPECIFICATION (normally aspirated) FAST ROAD			
	1585 (131)	1756 (124 Sport)	2*l* (131)
CR	9.6:1 (97 octane fuel)		
Pistons	Cast		
Fuel system	40 DCOE (32ch)	45 DCOE (36ch)	45 DCOE (36ch)
Fuel pump	Standard or Facet Silver Top		
Cams/pulleys	Standard or 30mm Flangeless with 1" belt		
Springs	Standard		
Valves	Standard, modified inlets		
Valve caps	Standard		
Head prep	Fully ported, blueprinted seats / valves		
Lubrication	Beta types except Monte Carlo – race sump – others standard		
Oil cooler	13-row ½ BSP 235mm matrix		
Crankshaft	Tap/plug, modified oilways		
Flywheel/clutch	Lightened. Standard clutch		
Con-rods	Standard		
Head gasket	130 TC		
Head bolts	Fiat late-pattern or GC race		
Balancing	Crank/flywheel only		
Plugs	Standard		
Ignition	Standard		
Exhaust	Standard or 4-2-1 preferred (4-1 can be used, but 4-2-1 will give better bottom-end torque)		
Output	126bhp @ 6600rpm	140bhp @ 6500rpm	155bhp @ 6300rpm
NOTES	An economical, flexible engine which capitalizes on the extraordinary torque output of the standard-cam/twin-carb set-up		

STAGE II			
	1585	1756	2*l*
CR	10:1		
Pistons	Cast		
Fuel	98 octane (or 50% 98/97 oct)		
Fuel system	40 DCOE (34ch)	45 DCOE (36ch)	45 DCOE (38 or 40ch)
Cams/pulleys	St II with adjustable/vernier pulleys, 1" belt		
Springs	Triple		
Valves/caps	As Fast Road		
Head preparation	As Fast Road with inlet port sizes to suit		
Lubrication	Race sump (except Monte Carlo, 105/130TC types)		
Oil cooler	13-row	16-row	16-row
Crankshaft	As Fast Road		
Flywheel/clutch	As Fast Road, uprated organic clutch		
Con-rods	Standard		
Head gasket	As Fast Road		
Head bolts	As Fast Road		
Balancing	As Fast Road		
Plugs	Road/competition NGK B9EGV		
Ignition	Standard		
Exhaust	Rally cams 4-2-1, race cams 4-1		
Output	132–145bhp @ 7500–8000	140–160bhp @ 7200–7800	172–185bhp @ 7000–7200
NOTES	An engine competitive at clubman level for race/rally/hillclimb etc, also suitable for kitcar road/race. Actual bhp will depend on standard inlet valve size (42/43½mm), cc, cam type. (St III cams may be used as the St II CR is high enough.)		

Assessing the condition of a TC for modification

If it is intended to modify an engine without a major stripdown, the following main inspection sequence should be carried out (ideally read the whole book first!):

Visual examination

1 Check the condition of the crankcase breather and inside the camboxes. Chronic oil neglect is a killer, and if this is the case, both items will show evidence of heavy carbon, varnish and sludge deposits. Suspect worn guides, damaged bearings, worn oil pump, rings, bores and crank, valve/ring and combustion chamber deposits.
2 Inspect the coolant galleries in the block (remove the water pump) or check the condition of the coolant. Presence of white emulsion may indicate a blown head gasket or cracked head/block. It will be readily apparent if antifreeze has been used regularly as there will be no evidence of corrosion. Use of plain water as a coolant can cause heavy sludge build-up, especially in the radiator and at the back of the block.
3 Check the condition of the cam belt. Do not crank the engine if it looks worn and due for replacement. Check the cam timing marks relative to TDC.
4 Examine the condition of the inlet/exhaust ports and valves (a torch may be needed). Heavy fouling of the exhaust ports may indicate worn rings, and similarly, if the valve guides/seals are worn, there will be heavy deposits on the valves (this is easier to see on the inlets). Fouling of the inlet ports indicates poorly seating inlet valves.
5 Look for signs of coolant/oil leakage around the block/head joint, which may indicate a blown gasket. Similarly, inspect the seals on the crank, auxiliary driveshaft and cams (damaged seals can indicate over-heating). Also check the core plugs.
6 Turn the engine over slowly by hand, with the plugs out, to ensure that when it is cranked, 'mechanical contact' does not take place.

Compression test

This is a very good way of assessing the condition of the rings/bores and valve sealing prior to stripdown. Remove the plugs and the main power feed lead to the ignition pack (or ground the main HT lead from the ignition pack to earth). Using a suitable (accurate) compression gauge, crank the engine several times,

with the throttle wide open, until the cylinder pressure reaches the peak attainable level. If the pressure comes up slowly, valve leakage may be a problem; a cold engine in good condition will produce a pressure of 8–10bar (117–147lbf/in²) almost immediately, and the pressure will continue to rise, more gradually, to 12–15bar (176–220lbf/in²). [1bar = 14.7lbf/in².] The peak pressure will depend on the cam type. Standard cams and short-duration competition cams will give readings at the upper end of the scale. Race cams and lower-compression engines, *eg* turbo, Vx engines, will give lower figures. Hot engines, particularly if the lubrication system is working, will give higher readings still.

If the compressions are low, squirt some engine oil into the low cylinders and crank the engine a few turns with the plugs replaced to disperse the oil over the rings. This will simulate rings/bores in good condition by improving their seal. Remove the plugs and test again – the compressions should go up (unless the inlet cam timing is badly retarded, *eg* one tooth out), indicating that the rings/bores are worn. An engine in good condition should give readings within 10% of each other. This test will indicate the need for 'head-off' or block stripdown. A low wet-compression result may indicate bent/damaged valves, blown gasket, holed piston, etc.

Head gasket

Models including the Vx, 130 TC and later have either the Goetze Astadur head gasket (brown in colour) or Fiat/Lancia's own (black or brown) polymer type. These types are immensely strong, and if the engine shows good compressions and there are no visible leaks from the head gasket, it may reasonably be assumed that it will withstand the extra cylinder pressures which will result from special tuning. GCT strongly advise that the early head gasket be replaced as a matter of course.

'Sump off'/oil pressure

(See also chapter on lubrication systems for analysis of sumps.)

Clearly a measurement of oil pressure prior to stripdown will give a good idea of the condition of the crank/bearings. Unfortunately, not all TC models had oil pressure gauges – and those that were fitted are not overly sensitive! A capillary-type gauge (Smiths, Stack, Raceparts are three of the best) can be connected to the engine for this test. A cranked engine should give 25–40lbf/in² cold; remember that if the engine has sat for a long period, especially if the oil has been drained, it

STAGE III			
	1585	1756	2l
CR Pistons Fuel	Ideally 10.5–11:1 minimum Forged 98 octane		
Fuel system	45 (36ch)	45 (38ch)	45 (40ch)
Fuel pump Cams/pulleys Springs	Facet Silver Top St III or St IV pulleys/belt as St II Triples		
Valves/caps	Alloy caps with 43½/36 or 44/36	44/45 in 36/38 ex	46 in 40 ex
Head preparation Lubrication	As St II, but larger in/ex port sizes. Modified coolant galleries As St II with Accusump or dry-sump (especially circuit/oval)		
Oil cooler	16-row	16-row	19-row
Crankshaft Flywheel/clutch Rods Head gasket Head bolts Balancing Plugs Exhaust	Stress-relieved, Tufftrided + as St II Steel flywheel, uprated or 7¼" clutch Lightened (shot-peening desirable) As St II As St II Crankshaft, flywheel, clutch, con-rods As St II As St II		
Output	145–170bhp @ 7800–8200	160–180bhp @ 7400–8000	185–200bhp @ 7200–7600
NOTES	An engine competitive in all forms of motorsport at National level. For 'full race' (St IV), increase valve sizes, raise CR; 'wilder' cams can raise peak bhp at expense of mid-range torque.		

TURBOCHARGED SPECIFICATIONS (This is by no means an exhaustive list, but covers the main types seen at GCT)	
1600 Delta (carb)	Heat-treated crank (as St III n/a) standard rods Steel flywheel 130TC cams, triple springs Fully ported head, standard valve sizes (modified inlets) 130 TC head gasket, 12.9 head bolts Cast pistons 7.8:1 CR Standard turbo / intercooler, 18lbf/in² boost Raised fuel pressure 180bhp @ 8000rpm, 170lbf ft torque 2l Gp N clutch cover, sprung sintered plate
1600 Delta *ie*	As above but with: 44 inlet valves 20lbf/in² boost (about the most from standard turbo with 130 TC cams – 7.8:1 CR) Remapped ECU (injector duration to maximum – standard ignition) 16v Integrale intercooler Uprated cover (20% more clamp load than Gp N) with sprung sintered plate) 230bhp @ 7800rpm, 210lbf ft torque (Detection Techniques rolling-road figure) Above this level, forged pistons, larger injectors, longer-duration cams plus larger turbo required
GpA 8v Integrale	Race 43½ inlet valves GpA cams, injectors, EPROM, fuel pump, regulator 27lbf/in² boost, copper head gasket, 'O'-ringed block Fully ported head Forged pistons, 7.5:1 CR Clutch as Delta 1600 *ie*, but 230mm dia 310 bhp @ 7200rpm, 300lbf ft torque
NOTES	The standard 16v Integrale gives about the same output with the same boost level using Gp A cams with the standard head

may give no pressure at all – and the oil pump will need to be primed by injecting oil into the feed gallery from the pump to the filter. An engine started from cold should readily extinguish the low-pressure warning light, not knock or rattle, and produce at least 15lbf/in^2 per 1000rpm, dropping to 10psi per 1000rpm hot, with 14–25lbf/in^2 at tickover. (*Note:* A persistent knock, if the compressions are good, can be caused by an incorrectly timed auxiliary driveshaft on models of 79.2mm stroke or longer. Other causes are a broken valve guide, bent valve, loose tappet clearances and worn cam, damaged bearings or worn out small-end bushes. If a bearing is badly damaged, the pistons may hit the head.)

Models with a good sump design, especially if fitted with an oil cooler, which have been well maintained may have perfect crank/rod/bearing/oil pump assemblies – *ie* a good 'bottom end'. However, GCT recommend that, particularly on lesser models, the sump should be removed and the bearings inspected – simply unbolt the bearing caps, inspect and replace one at a time. Do not dislodge the bearings from their housings if they are in good condition.

Very minor marking can perhaps be tolerated, but if the bearings show signs of breaking up or severe scoring/wear, strip the bottom-end down and prepare properly.

Inspection of the oil pump pickup is another way to determine the history of an engine (broken bearings etc will be trapped in the strainer), and while the sump is off the pump clearances can be checked.

If the big-end bearings are good, the mains will probably also be fine. Check that the thrust washers are in place and intact.

Author's note: Nearly every TC that GCT bought from a scrapyard was just that – scrap! Some of the worst faults were:

- cam dowels removed so that the engine turned over fine! When the head was removed, all the valves were found to be bent.
- worn out thrust washers. The engine appeared visually OK, but there was massive crank end-float. The total collapse of the thrust washers had caused the crank webs to impact on the rods and main bearing housings. The whole block was scrap.
- water, corrosion and broken glass(!) in the cylinders. The engine had been stored on its side with the inlet manifold removed.
- all four plug threads cross-threaded.

Conclusion: Very unwise to use an engine from a scrapyard in a car without fully checking it first!

Specification list

Having identified a suitable (race-legal) engine/car combination, the next step is to calculate an outline budget for the engine so that a combination of power and reliability can be achieved which will make the vehicle both competitive and cost-effective. Ultimately, the power unit must be 'integrated' so that a mismatch of components (*eg* carbs too big, CR too low, clutch too weak) can be avoided. (Ideally, the companies approached should be able to supply accurate torque/power predictions with their components.)

Break the engine down into its component parts as follows (it may help to photocopy sections A-F): [*Author's note:* no infringement of copyright will result!]

A – BLOCK PREPARATION	Notes	£
Decarbonize, chemically clean and paint		
Reface	depth	
Rebore/hone/chamfer	size	
'O' ring, etc		
Auxiliary driveshaft bearings		
Modify auxiliary driveshaft		
Supply pistons (c/w rings, pins, etc) (forged/cast)	size	
Modify pistons (dome, cutouts, bowl volume)	CR reqd	
Inspect, resize con-rods as required		
Replace bushes		
Con-rod mods (lighten, shot-peen, etc)		
Con-rod nuts and bolts		
Con-rod balance		
Piston balance		
Additional items (tensioner, core plugs etc)		
B – CRANKSHAFT / FLYWHEEL		
Inspect crank		
Tap and plug		
Stress relieve		
Regrind/lap (polish)		
Tufftride		
Modify oil ways		
Reface flywheel		
Lighten flywheel		
Modify flywheel (for alternative clutch)		
Ring gear		
Steel flywheel		
Clutch type (standard type/7¼"/5½")		
Dowel flywheel to crank		
Balance: Crankshaft		
Flywheel		
Clutch		
Front pulley		
Flywheel bolts		
Release bearing (may need mod to 7¼" clutch)		
Bearings: M		
B		
T	sized to crank	
C – HEAD PREPARATION		
Decarbonize, beadblast, clean		
Reface	depth	
Valve guides		
Recut seats (standard/competition)		
Recondition valves		
Modify valves		
Big valve inserts		
Unleaded ex valve seats		

Race (or larger) valves (sodium-cooled ex?)		
Porting (gasflow)		
Flowbench test		
Valve spring type		
Valve spring seat relief		
Spring seats		
Valve caps		
Cam buckets		
Shims (possible machining of cam carriers)		
Cam type		
Line boring for oversize cams		
Cam pulleys, belt		
Inlet manifold type		
D – FUEL SYSTEM		
FUEL INJECTION		
Fuel injection system (non-standard)		
Injector type		
Fuel pump		
Regulator		
Fuel filter		
Hoses		
ECU		
(Systems such as Weber Marelli include throttle bodies, ECU, sensors, injectors, etc – enquire prior to purchase)		
CARBURETTORS		
Carburettor type/size		
Jets/chokes/levers/T-piece		
Rampipes		
Linkage		
Fitting kit ('O' rings, etc)		
Fuel pump		
Fuel filter		
Regulator		
Air filter		
Airbox, ducting		
E – LUBRICATION & COOLING		
Sump conversion		
Lubrication accessories (remote filter, sandwich plates, hose, etc)		
Oil pump and gear		
Dry-sump system		
Oil filter		
Cooler/heat exchanger, ducting		
Accusump		
Turbo oiler		
Radiator, ducting		
Water pump, incl drive layout		
Thermostat, header tank		
Coolant pipes/clips		
Catchtank		
F – TURBOCHARGING		
Turbocharger type		
Wastegate		
Dump valve		
Intercooler (high boost may need a stronger type)		
Water spray (to intercooler)		
Knock sensor		
ECU type		

Hoses/cabling/clips		
(For 'blow-through' operation with carb fuel system, sealed float chamber required)		
Turbocharger overhaul/modification (compound turbo?)		
Adjustable or two-stage boost control		
Supercharging		
G – EXHAUST SYSTEM		
4-1 or 4-2-1 (to appropriate lengths/diameters)		
Silencer (a single box is preferred)		
Brackets and pipework		
H – IGNITION SYSTEM		
Plugs (running-in/race)		
Leads		
Distributor components		
Rev-limiter		
ECU		
Crank sensors, 'chopper'		
Coil/power pack		
Battery (check RAC requirements)		
J – MISCELLANEOUS ITEMS		
Head gasket/sealing system		
Other gaskets/seals		
Plated parts		
Sundry fasteners		
Head bolts		
Instruments and fittings		
Oil (running-in/race)		
Antifreeze		
Rolling road or dyno test, mapping		
Outright engine parts purchase		
Strip/build cost		

Added to the above, the cost of a suitable gearbox should be considered, plus ancillary items, *eg* clutch, bellhousing, starter motor, alternator, engine/gearbox mountings, propshaft conversion/balancing, wiring, fuel tank and secondary hosework.

It is worth fitting race valve springs and uprated clutch if a 'cam swap' is envisaged at a later date. Similarly, if high-rpm cams are likely to be fitted later, fit forged pistons at the outset.

Following a thorough inspection of the stripped unit, it is usually possible to make savings: never commit yourself to a rebuild until a 'strip/inspect/report' has been carried out.

Note: Beadblasting of certain components, *eg* 16v head, race sump pans, is not advised due to the difficulty in removing media during cleaning.

CASE HISTORY No 1

Owner	GCT
Engine No	GC 210
Type	Fiat 2091 (86mm bore)
Use	Road-race
Tested	Warrior Automotive, Aug '94
Rig	Superflow

This engine was built for use as a demo engine and was used initially by Midtec Sports Cars in a Midtec Spyder, proving both reliable and extremely fast. The spec comprised:

Forged pistons 11:1 CR.
Fiat 130 TC valves, 43.5/36, triple interference springs.
IIIA cams.
Baffled 'big-wing' sump with Accusump.
GC offset sidedraught manifold, 45 DCOE (38mm choke) carb.
Marelli electronic ignition (block-mounted distributor).
Facet Silver Top (competition-spec) fuel pump.
NGK B9 EGV plugs.
The following tests utilized a 22" 4-1 exhaust manifold, though a 4-2-1 would have been preferred.
The engine was run-in for 2½ hours and then compression-tested (hot, wide open throttle)

1	2	3	4
205	195	210	210lbf/in^2

Ignition was set at 36° @ 5500rpm

TEST 1 Cams timed at 110°

SPEED (rpm)	TORQUE (lbf ft)	POWER (bhp)	BSFC (gm/kW hr)	
4000	135.8	103.5	409	JETTING: 160 main 175 air
4500	141.3	121.1	388	
5000	140.7	134	353	
5500	145.9 max	152.9	324	
6000	142.0	162.3	340	
6500	138.6	171.6	333	
7000	134.3	179.1 max	328	
7500	120.5	172.1	349	

Results: In order to lean out the mid-range jetting the mains were reduced to 155, but to avoid excessive leanness at the top end the air correctors were also reduced to 170. The initial results were very encouraging, although the engine was a little rough at low revs; this would be dealt with later.

TEST 2

SPEED (rpm)	TORQUE (lbf ft)	POWER (bhp)	BSFC (gm/kW hr)	
4000	134.8	102.7	369	JETTING: 155 main 170 air 55 F9 idle
4500	144.7	124	358	
5000	139.2	132.6	343	
5500	148.1 max	155.2	316	
6000	143.9	164.5	320	
6500	138.8	171.9	320	
7000	136.3	181.7	309	
7500	125.3	179	326	

The test was immediately re-run from 4250–7250rpm at 500rpm intervals to examine the torque response more closely; idle jets were swapped to lean-out the bottom end.

TEST 3

SPEED (rpm)	TORQUE (lbf ft)	POWER (bhp)	BSFC (gm/kW hr)	
4250	146.2	118.4	359	JETTING: 55 F8 idle (bigger air bleed)
4750	138.5	125.3	358	
5250	147.8	147.8	317	
5750	147.5	161.6	306	
6250	142	169.1	314	
6750	139.1	178.8	325	
7250	131.5	181.6	332	

It was immediately evident that the changes of jetting had enhanced the bottom end and peak torque, with useful gains at 7000/7500rpm as well. On the flexibility test, the engine would hold full throttle down to 2250rpm; this characteristic coupled with the flat torque would give the engine excellent acceleration. Throttle response was excellent.

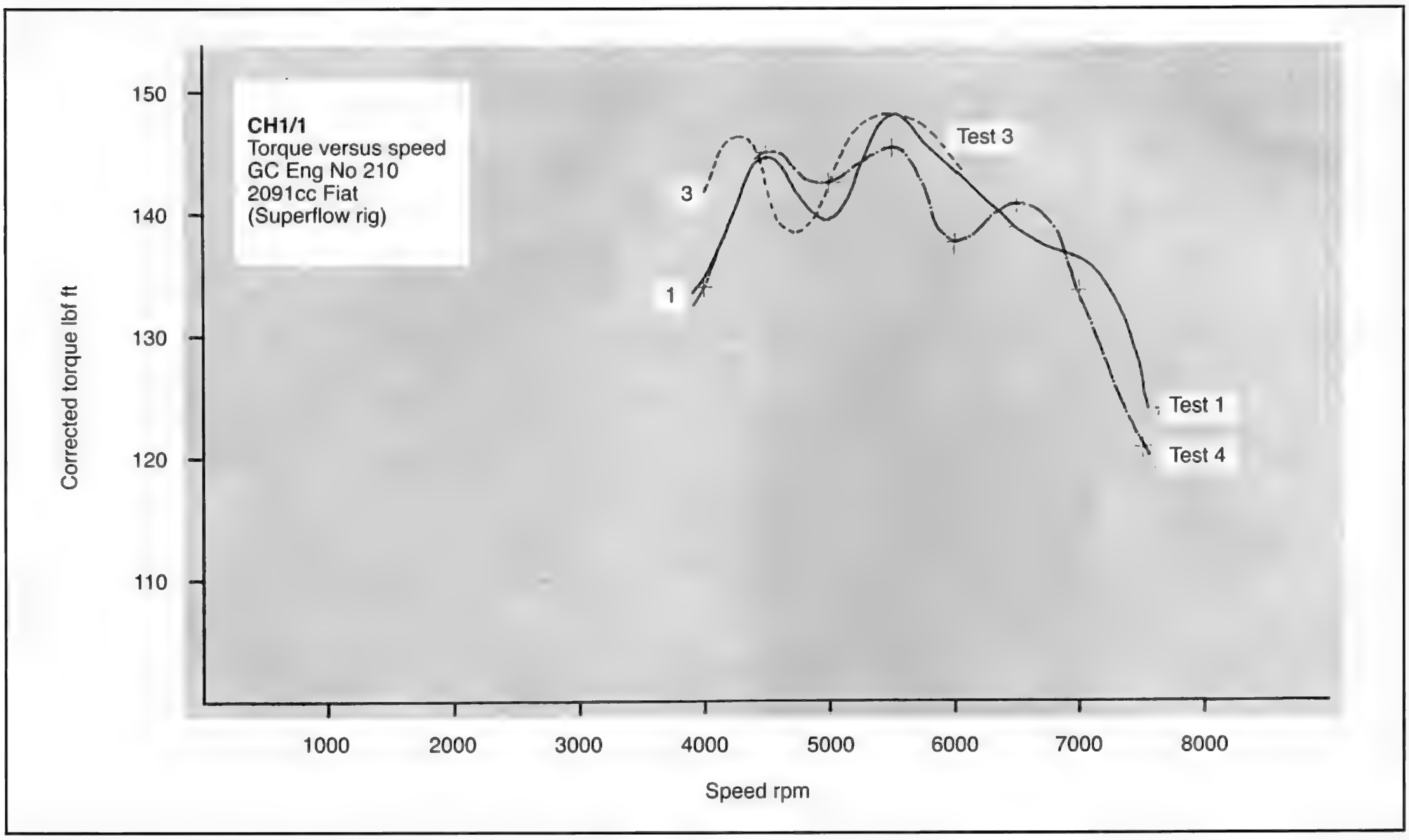
Next test involved testing the engine with the single silencer fitted to observe the effect of back-pressure on torque and jetting. The silencer was fitted approximately 2ft from the collector of the 4-1 manifold. This was an oval, straight-through design, 18" × 7" with 2" ID.

CASE HISTORY No 1

Fitting the silencer reduced torque between 5250 and 6250rpm although at this stage the jetting was over-rich, caused by back-pressure restricting the airflow into the engine. However, a useful increase in torque was achieved at 5000rpm (+3.1lbf ft) and 6500rpm (2lbf ft). This was probably due to the damping effect of the silencer on the pressure waves in the manifold. Changing the air corrector to 175 reduced the bsfc @ 7500 to 352, but in fact led to a torque loss as well (0.4lbf ft). The curious hump in the torque curve on all tests at 4500rpm was probably caused by reflected exhaust wave action rather than any inherent peculiarity in the engine!

TEST 4 (with silencer)

SPEED (rpm)	TORQUE (lbf ft)	POWER (bhp)	BSFC (gm/kW hr)
4000	134	102.1	388
4500	145	124.3	359
5000	142.3	135.5	341
5500	145.1	152	334
6000	137.5	157.1	316
6500	140.8	174.3	338
7000	133.7	178.3	327
7500	120.9	172.2	388



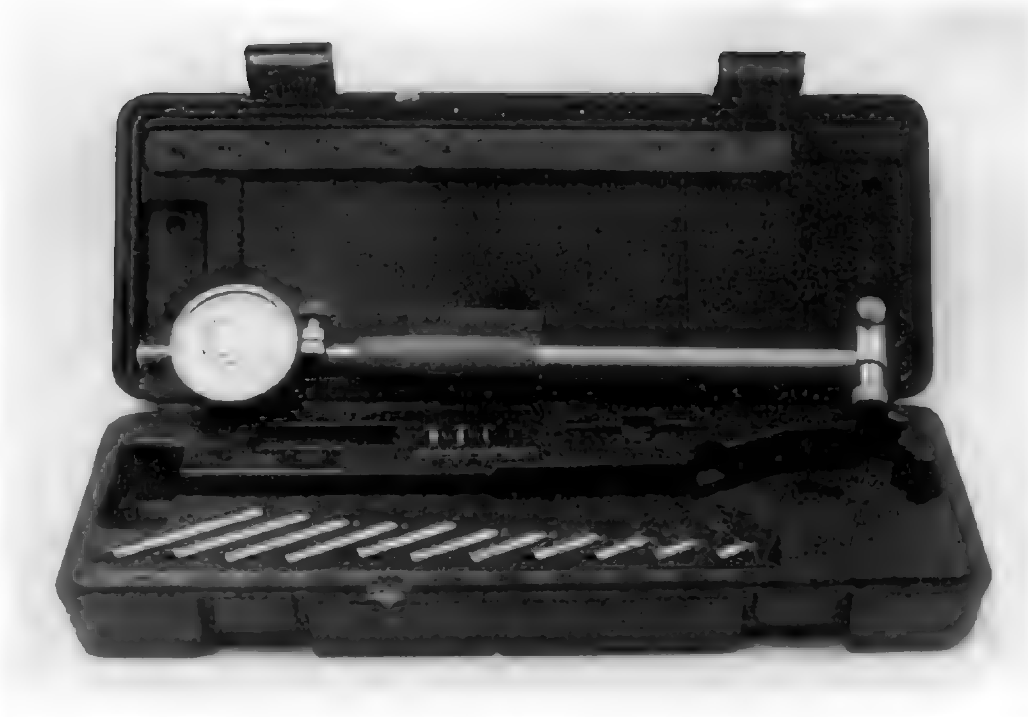
CHAPTER 3

TOOLS

One great advantage of the TC is the limited number of tools required to strip and rebuild it.

3/1: Magnetic base dial gauge 0–12mm necessary for finding TDC and timing cams, as well as several other uses. Digital micrometer 0–1" is useful for shim measurement. Other micrometers for checking crank, cams, pistons and setting bore gauge 24–50mm, 50–75mm, 75–100mm. Vernier calliper 0–6" has million and one uses, digital types are especially good (eg for checking con-rod big-end bores).

3/3: (Fiat tool.) Useful for fitting heavy wire circlips in piston grooves. Barrel of tool has tapered bore, compresses clips prior to fitting.



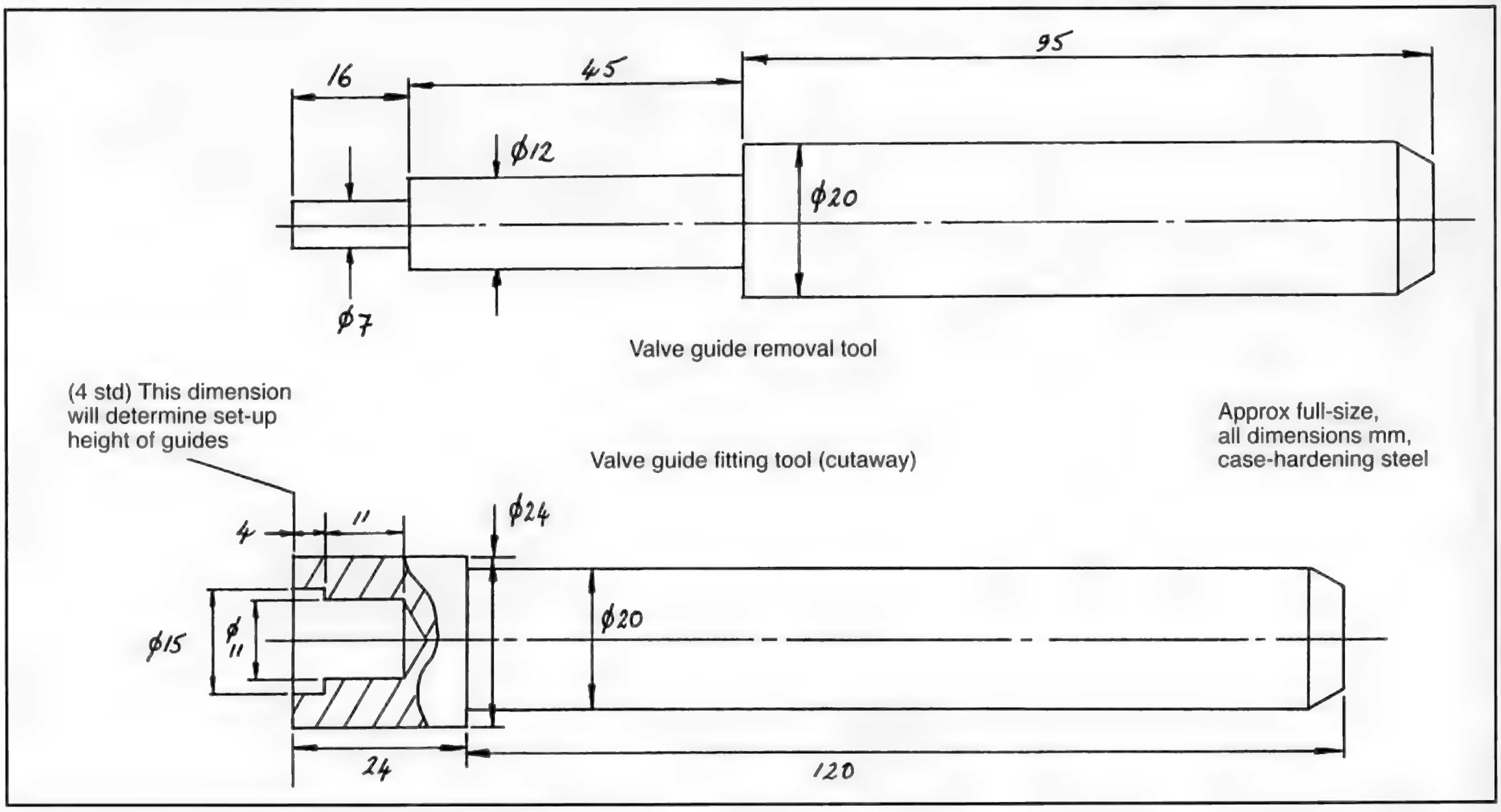
3/2: Bore gauge, essential for checking bore sizes in block, con-rod housing diameters. Gauge must be preset using micrometer; dial measures variations either side of setting. This model is Mitutoyo.

3/4: (Fiat tool.) The best way to change shims when head is already assembled. Holds bucket down so shim can be removed. Original tool for TC is hard to get but 1867055000 works equally well.

3/5: (Fiat tool.) Perfect tools for holding cam wheels and aux driveshaft pulley in place for bolt removal/fitting.

3/6: Piston clip fitting tool assembled.

3/7: Ring compressor. Piston fitting tool essential for compressing rings for fitment of pistons into bores.

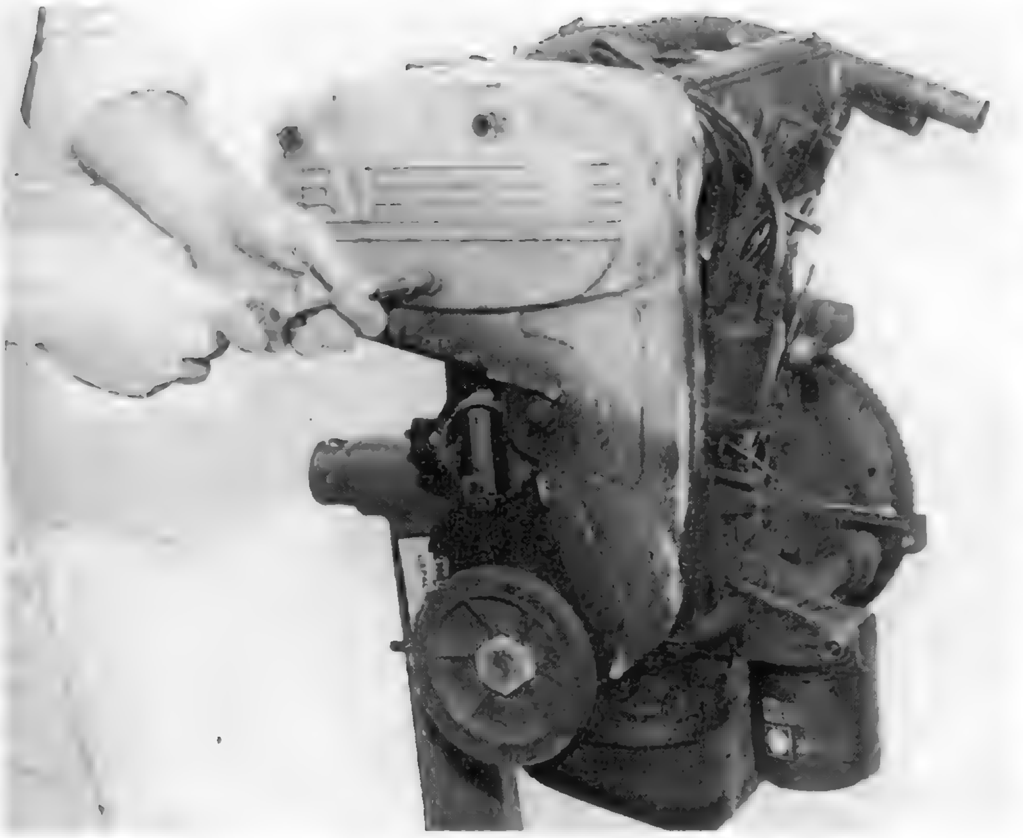


3/8: Valve guide tools (can be used with early or late guides).

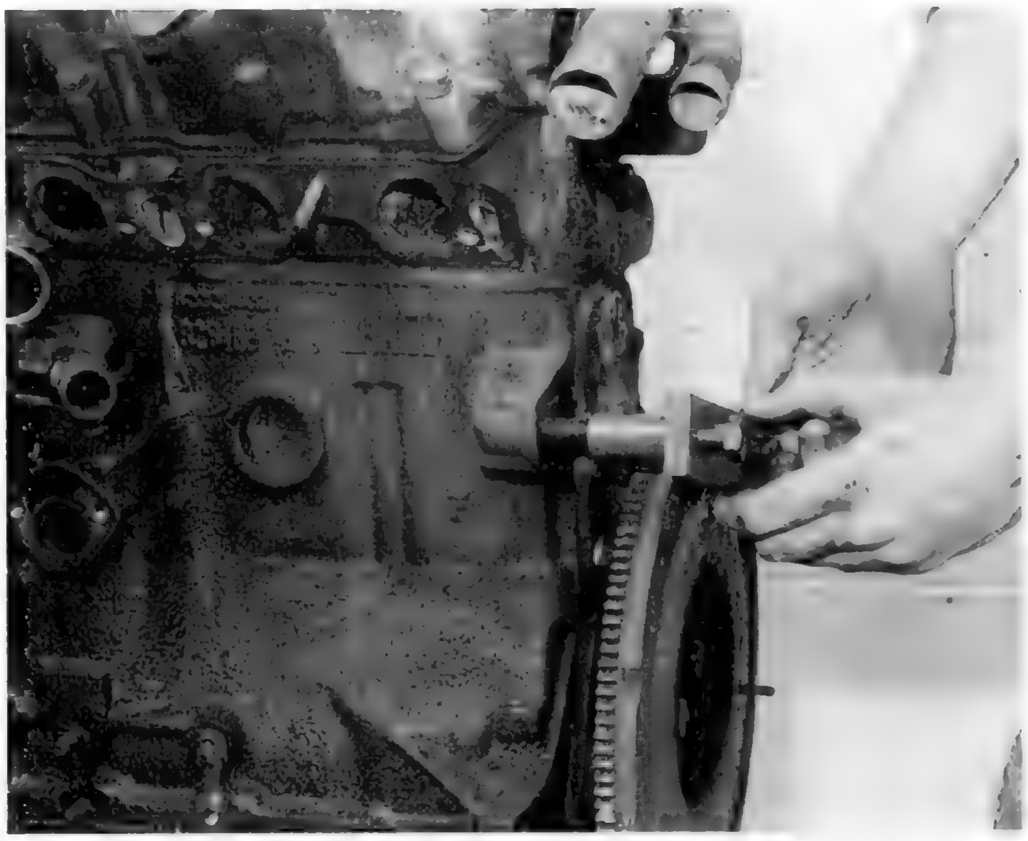
In addition to these particular items, the well-assembled tool set should contain the following:
¼" drive sockets 7–13mm (6 point drive).
⅜" drive sockets 10–19mm, spark plug sockets (6 point drive), angle drive.
½" drive sockets 17–24mm (preferably 6 point rather than 12).
A 22mm ½" drive socket is needed for models with reverse-thread crank front bolt, 38mm for the earlier nut type.
An air-driven impact wrench is especially useful (strip-down only).
Deep-drive 14mm/15mm ⅜" drive sockets should be used on rod nuts.
Late-pattern torque-angle bolts require a special tool from Fiat.
Feeler gauge.
Open-ended/ring spanners 10mm–19mm.
Ratchets to above sockets, extension bars and high-torque 'break bar'.
Accurate torque wrench (two may be needed) 16lbf ft–120lbf ft.
Long-nosed, narrow pliers (for removing shims).
Socket hex-head wrenches 5mm–12mm (sump plug, other socket hex bolts).
360° protractor approx 6" dia.
Adaptors ⅜" drive for sockets.
Solvents for cleaning are mentioned in the appropriate part of the text, but it is worth mentioning a couple of items required for the rebuild which are often forgotten! These are:
Graphite grease or Graphogen (for initial lubrication of bearings, pistons, cams, etc).
Silicon gasket (*eg* Loctite).
Loctite studlock/nutlock (for carburettor mounting studs, sump bolts, etc).
Red/white marker paint (a small touch-up paint bottle is ideal).

CHAPTER 4

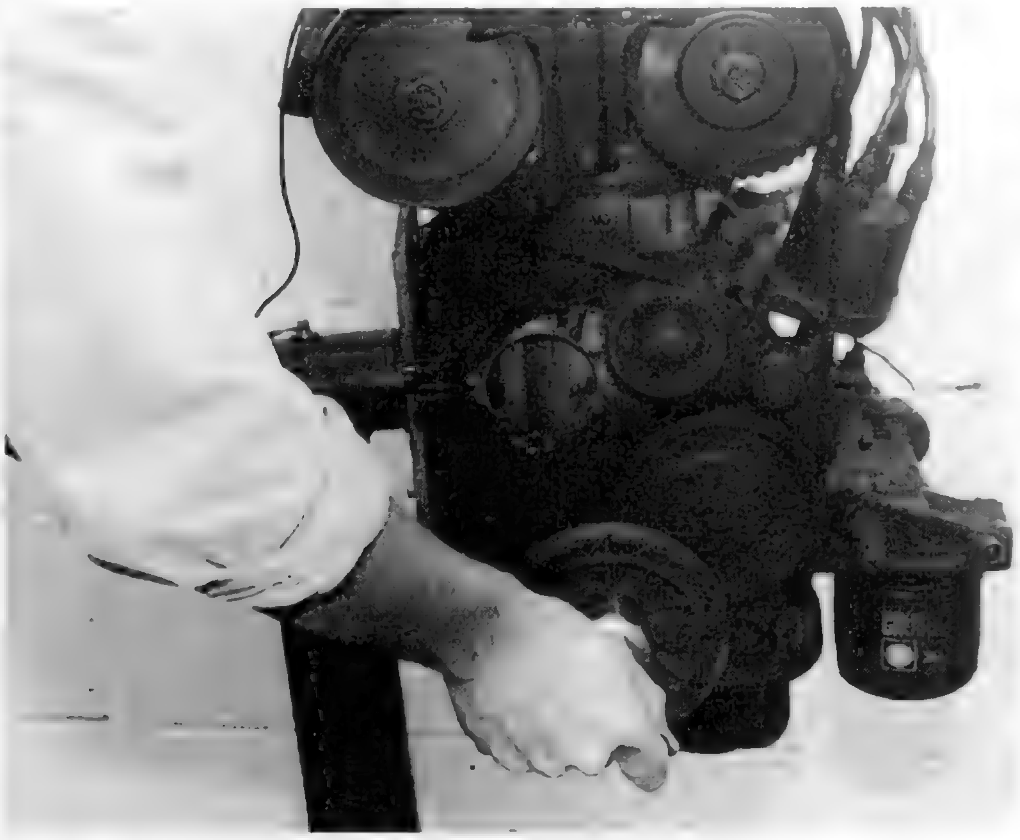
STRIPPING AND INSPECTING



4/1: Engine is 2l Lancia. Inlet and exhaust manifolds have already been unbolted. Remove 6mm bolts (10mm socket) which retain cam belt cover; don't lose rubber washers if fitted. Water pump already removed (6mm bolts). Note 20° rearward inclination used on all Beta-type installations visible from sump design. Keep the special short water pump pulley bolts.

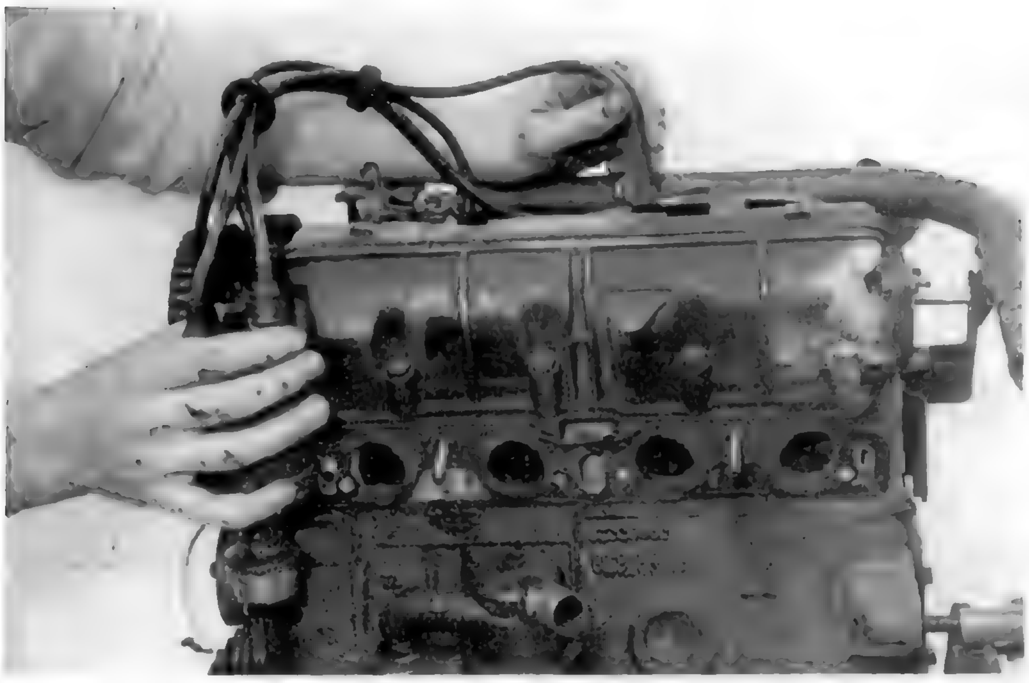


4/2: Lock flywheel with steel strap using bellhousing bolt to block (M12) and 8mm bolt to flywheel.

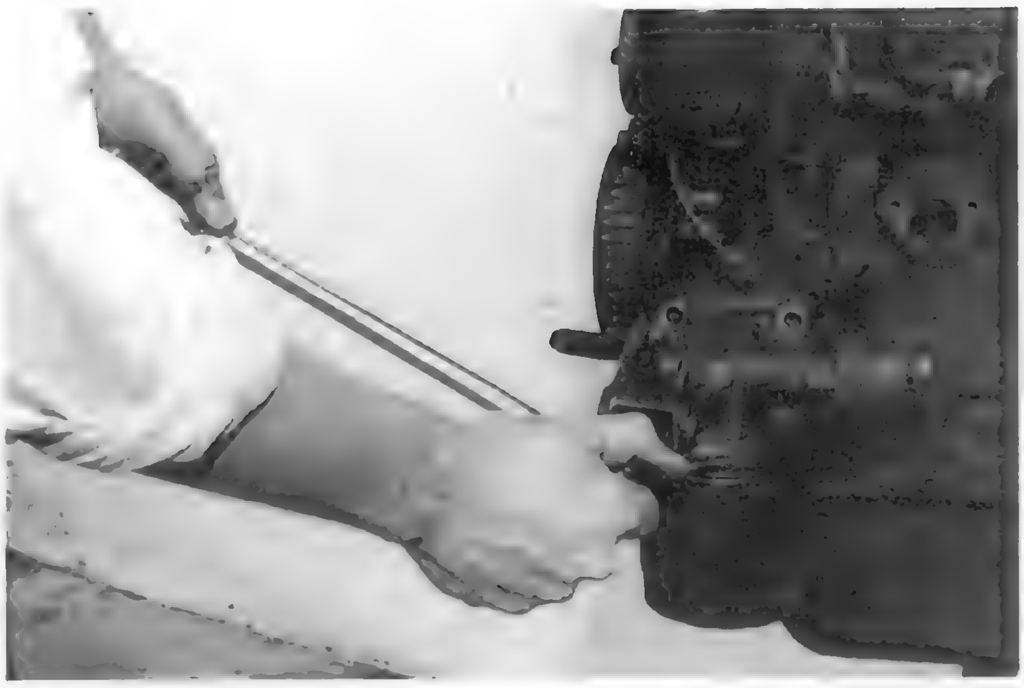


4/3: With flywheel locked, undo crank front pulley nut (38mm socket). Some early and all late models have reverse-thread bolt (22mm socket).

4/4: Provided crank is not turned so as to clash with valves, cam belt may be safely cut and removed. Note plastic (right) and steel (left) flanged cam wheels; rear flange is inlet, front flange exhaust. Top rail for cooling circuit visible between cam boxes. Cam timing marks line up with pointer when 1+4 cyls are at TDC (4 firing).

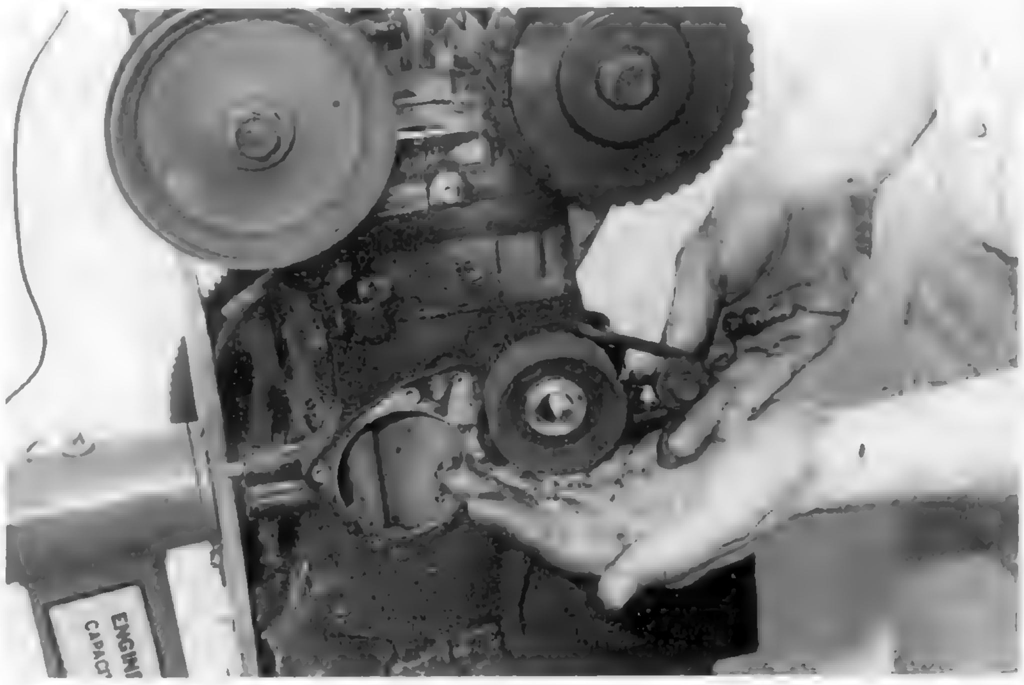


4/5: Remove distributor and HT leads. Distributor is driven off auxiliary driveshaft on this model via spline drive in driven gear, useful since it can be refitted in any position. Note coolant galleries in centre of head face and at ends for heating standard manifold.

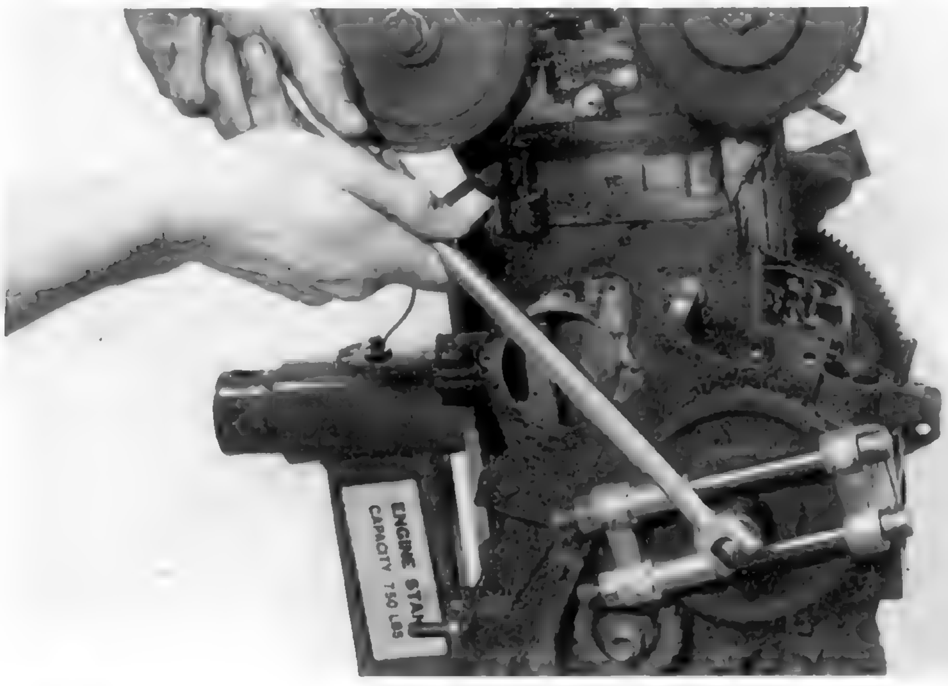


4/6: Fuel pump removed (13mm socket). Oil filter housing retained by four 10mm dia bolts (17mm socket). Remove oil pressure/temperature senders prior to cleaning. Housings vary from model to model. Lancia Beta type also carries alternator.

4/7: Loosen breather separator (13mm socket) and clips on rubber pipe. Unit separates oil from vapour for return to sump. Vapour and crankcase gas is bled to air filter. Note that bolt is sealed with aluminium or copper washer which should be replaced when refitting. On poorly maintained high-mileage engines, internal condition of separator will be heavily clogged with carbon deposits. Keep hose – some models are no longer available from Fiat.

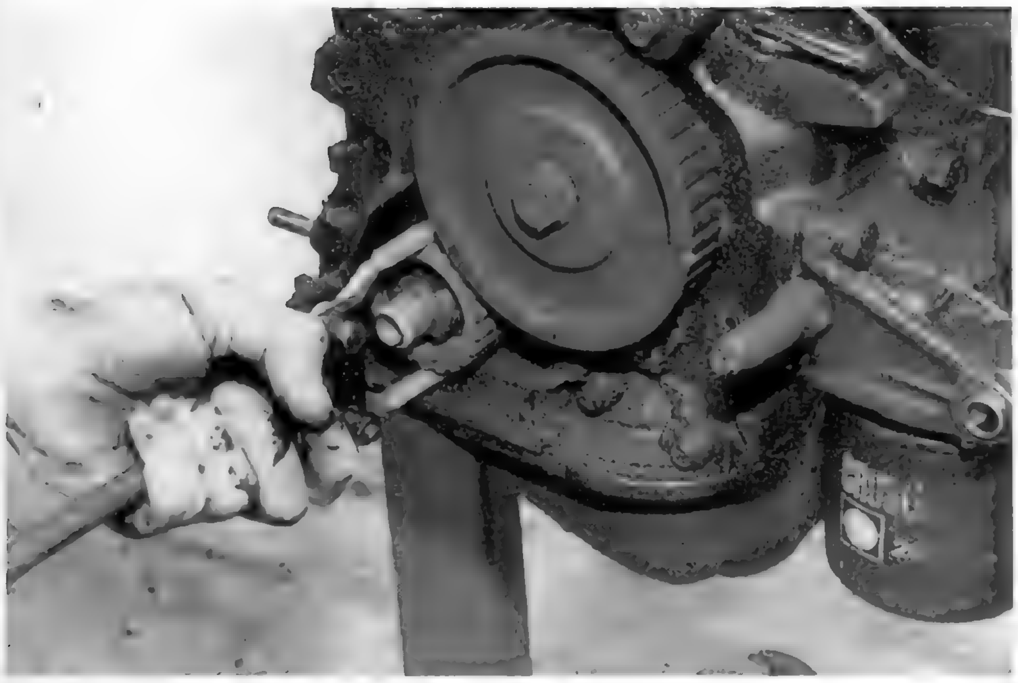


4/8: Slacken tensioner assembly (17mm/13mm socket) and remove. Keep special bolts which secure baseplate and spring, plus sleeve and washers. Beta tensioner bearing can be removed and replaced if needed; 131/132 type is integral with pulley. Note clean condition of cooling jacket. Late models use eccentric-type retainer with no spring (10mm Allen key).

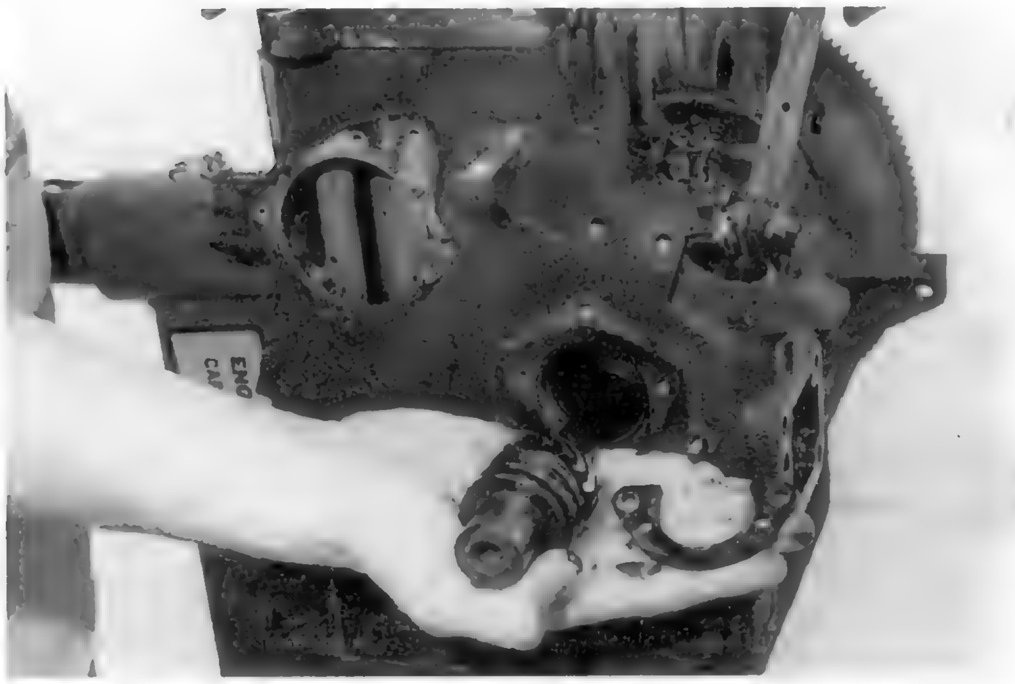


4/9: Fiat tool being used to secure auxiliary driveshaft pulley so bolt can be removed (19mm socket). If an air impact wrench is used, the pulley can be held by hand, but wear a glove to protect fingers.

4/10: Unbolt auxiliary driveshaft casing and remove (10mm socket). During cleaning process, make sure all traces of old gasket are moved. Later models dispensed with auxiliary driveshaft and drove oil pump off crank nose.



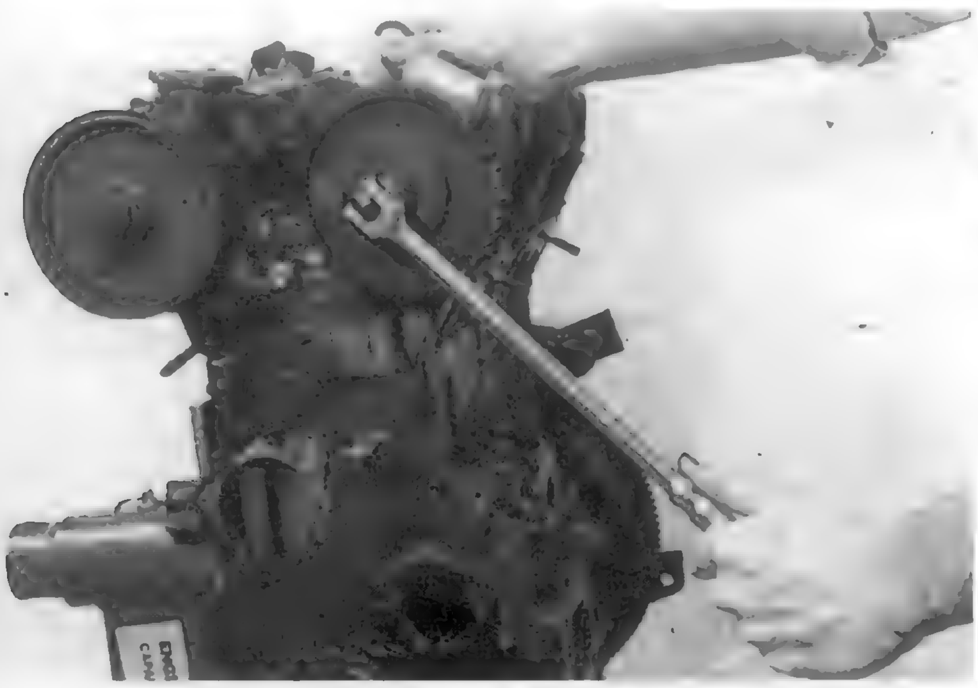
4/11: If nose pulley proves hard to remove, use a two-legged puller, or try two large-bladed screwdrivers to wedge it off once auxiliary driveshaft pulley has been removed.



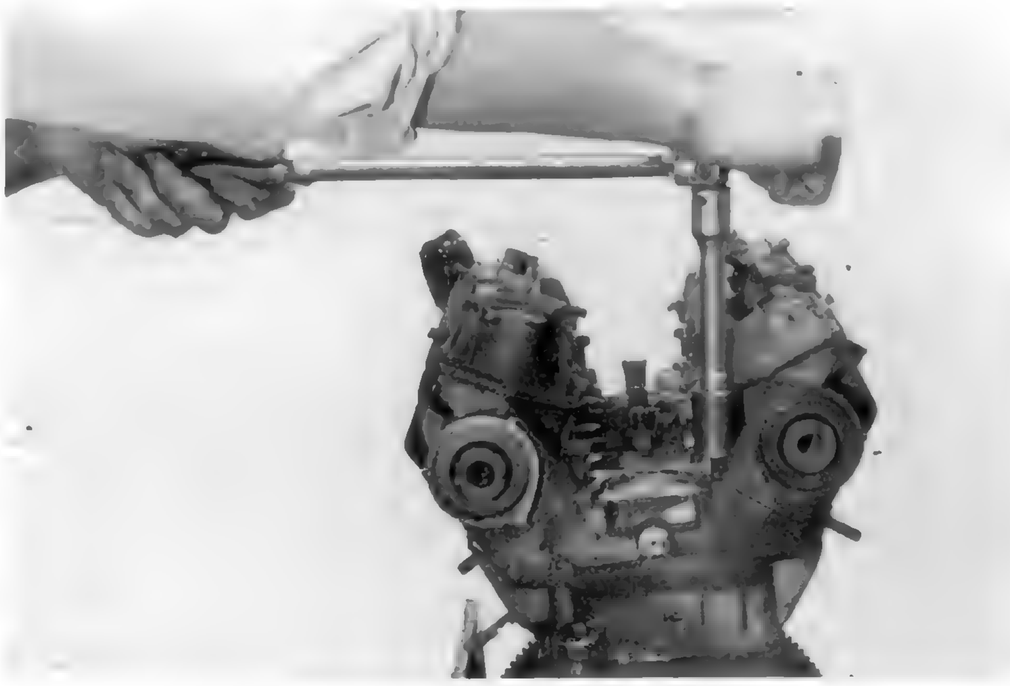
4/12: To remove auxiliary driveshaft gear, drive a small piece of wood into it and lift out. It is usually necessary to twist the shaft slightly. This is a useful tip when replacing a gear with worn splines without removing sump. Remove collar that secures shaft (10mm socket) and withdraw shaft. Inspect driveshaft bearings; if they are reasonable, leave alone. Replacement needs special tools and reamer and bearings are hard to obtain!

4/13: Auxiliary driveshaft and driven gear used on all early models. Check condition of spline inside gear – it drives oil pump. Note wear ridge on fuel pump lobe. Plug at RH end of shaft seals oil gallery along centre to feed rear bearing.

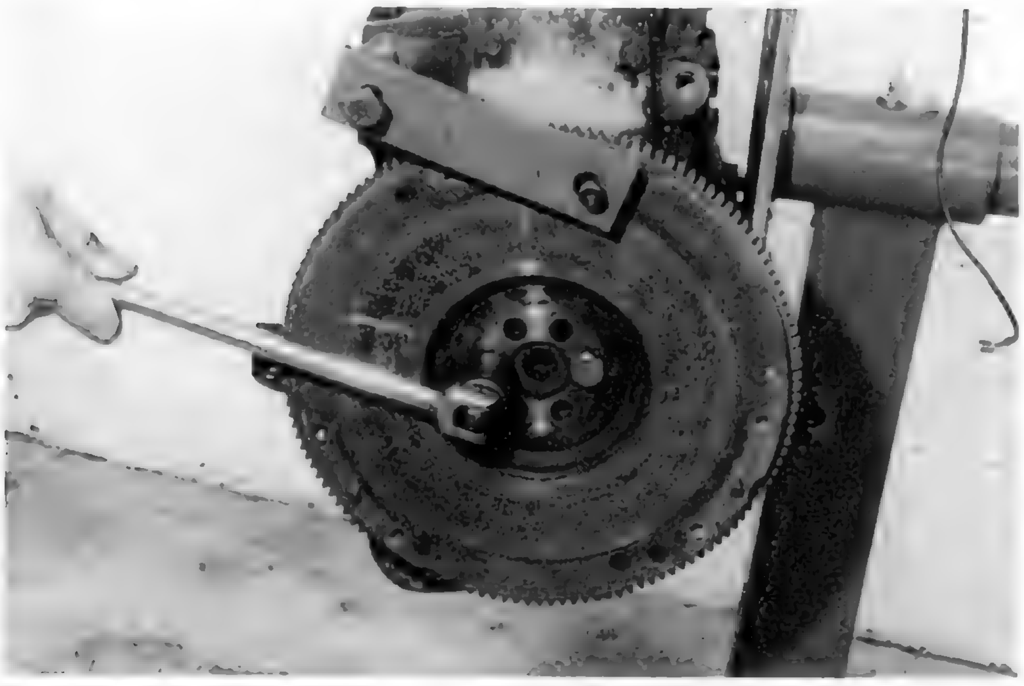
4/14: Check condition of these three parts. If tensioner feels loose or sticks, replace. Wear on pulleys can be severe if used in dirty conditions, eg rally or grasstrack. Worn nose pulley can upset cam timing. Check plastic pulleys for cracks, especially flanges.



4/15: Use cam wheel locking tool to undo bolts without turning cams (risk of bending valves). Alternatively, unbolt cam boxes and clamp pulleys in vice; don't damage flanges. Keep all three pulley bolts – they are 12-grade super high-tensile – and washers.

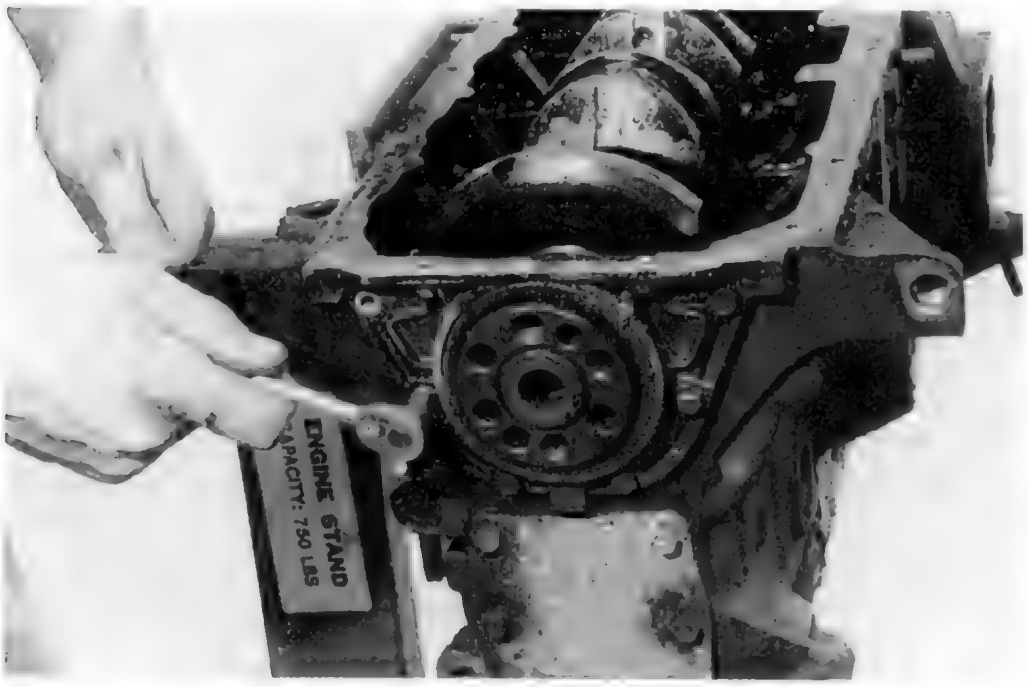


4/16: At this stage, head can come off. Remove head bolts starting from outer ends and working towards middle. Cam seals can leak on high-mileage engines; they can be replaced in-situ if required. Retain head washers. Stretch bolts (OE type) can be re-used up to three times, but if in doubt replace. Junk the early pattern bolts – they will be too old to be trusted!

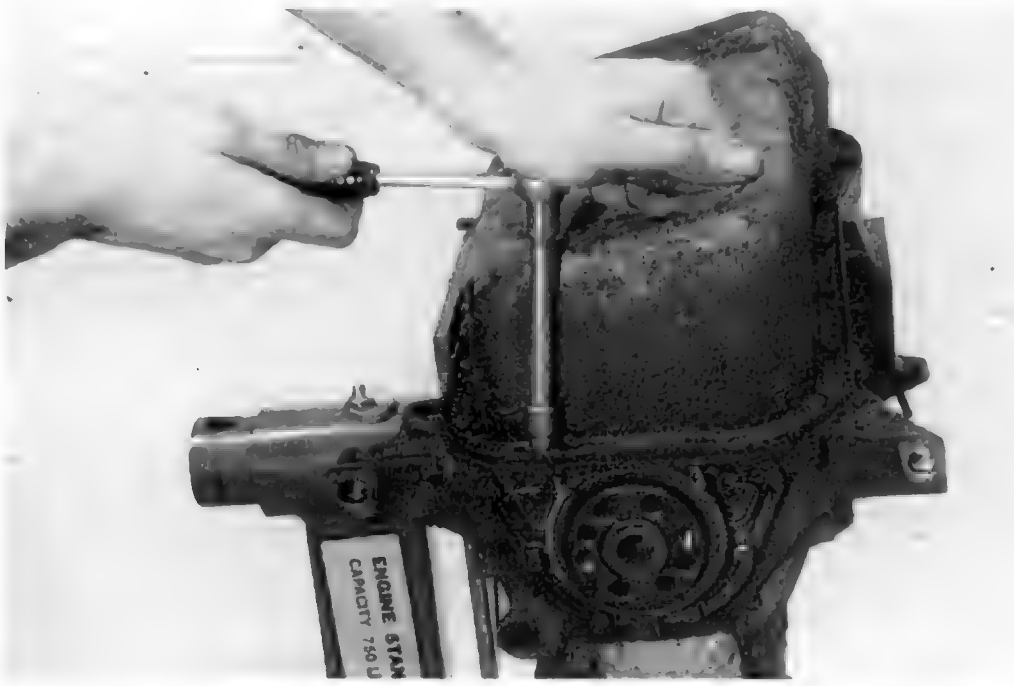


4/17: Loosen six flywheel bolts. Washer (not shown) under bolts must be retained; 1756 124 Sport and 1608, 1585 cranks have 10mm bolt (17mm socket), all others have 12mm bolt (19mm socket). Note bush in crank end – transverse engines do not require bearing. Bush centres flywheel on crank and must be retained. (If removing flywheel bolts with air tool, dampen clutch dust with aerosol oil first.)

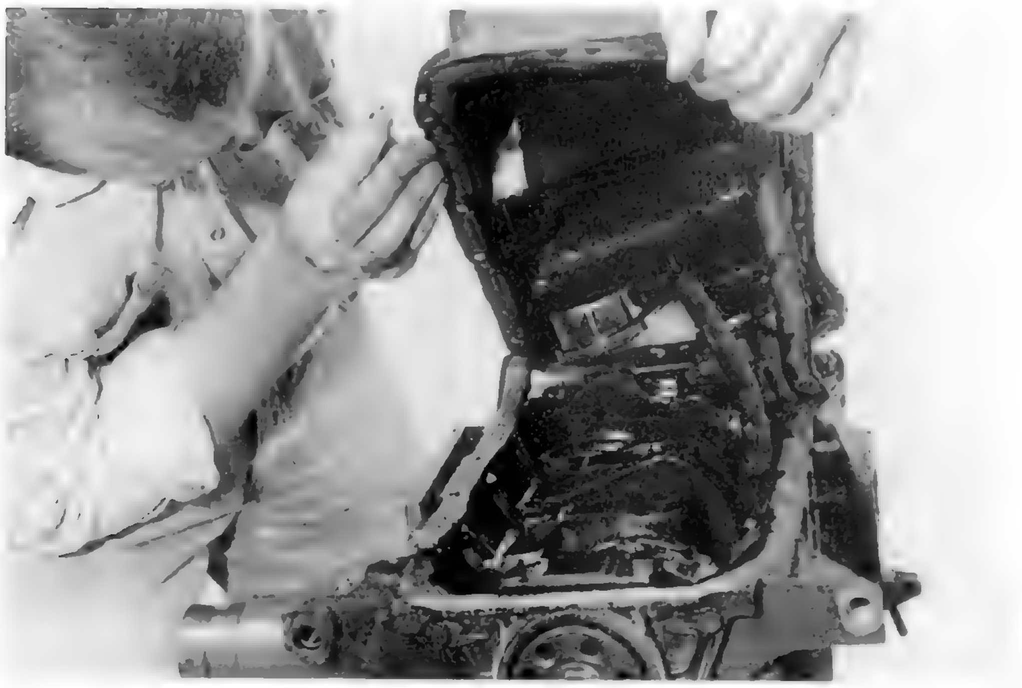
4/18: At this point block has been inverted to drain any remaining oil and coolant and give access to sump. Lift off flywheel. (Be very careful not to drop it when the last bolt has been removed!)



4/19: Undo six bolts (10mm socket) that retain crank rear seal housing and remove.

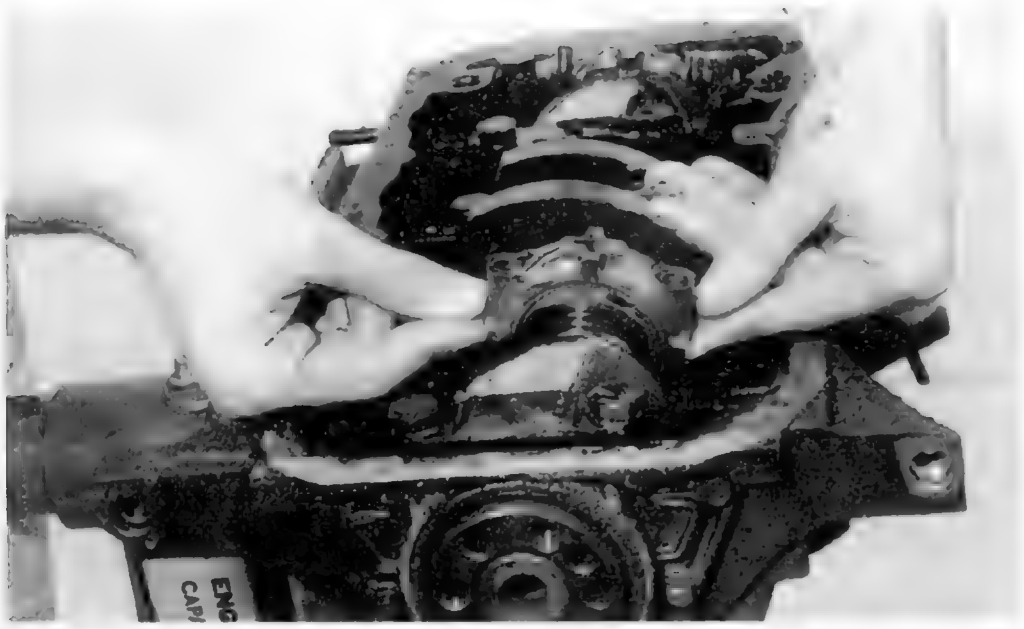


4/20: Slacken and remove 18 sump bolts (10mm socket). Late models, eg Integrale, have two-piece sump – alloy casing with steel baseplate, but fixing method essentially similar. Battered condition of sump base is usual!

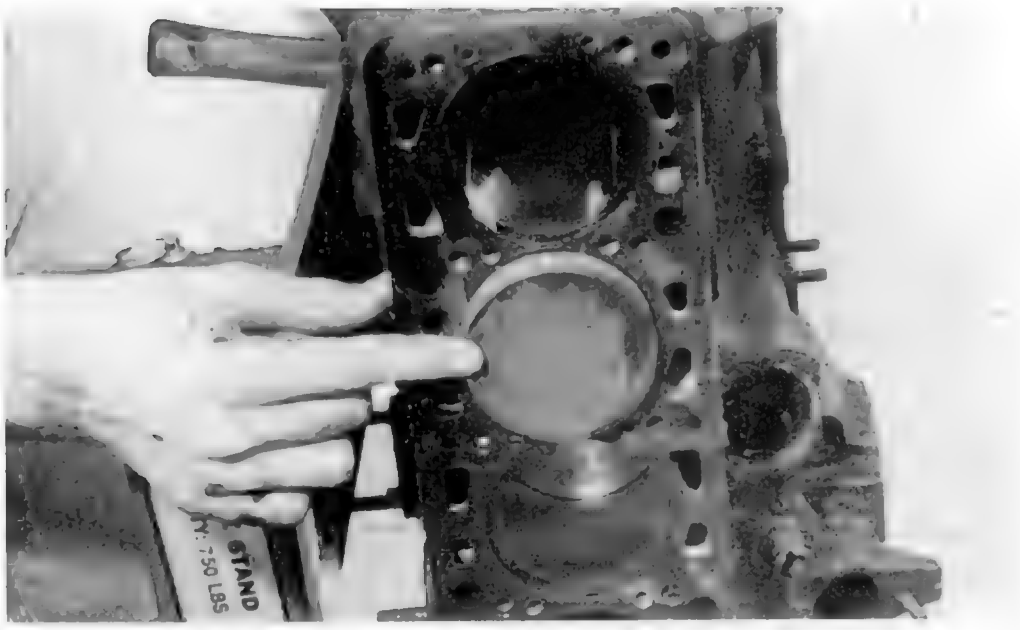


4/21: Sump may require use of bolster chisel or similar to free it; remember to straighten it afterwards. Oil pump and breather return pipe visible. Despite apparently poor condition of intervals, engine was in good reconditionable state.

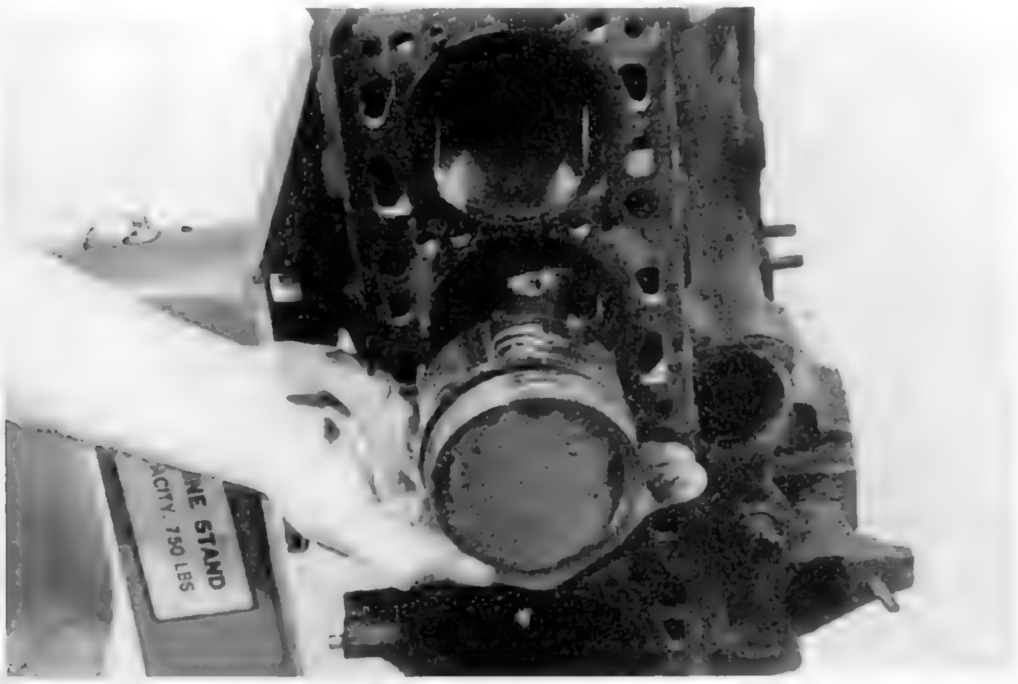
4/22: Loosen oil pump retaining bolts (13mm socket) and remove pump, plus two bolts (10mm socket) that retain breather pipe. Late models have front-mounted oil pump driven off crank nose; pump is contained in front seal housing.



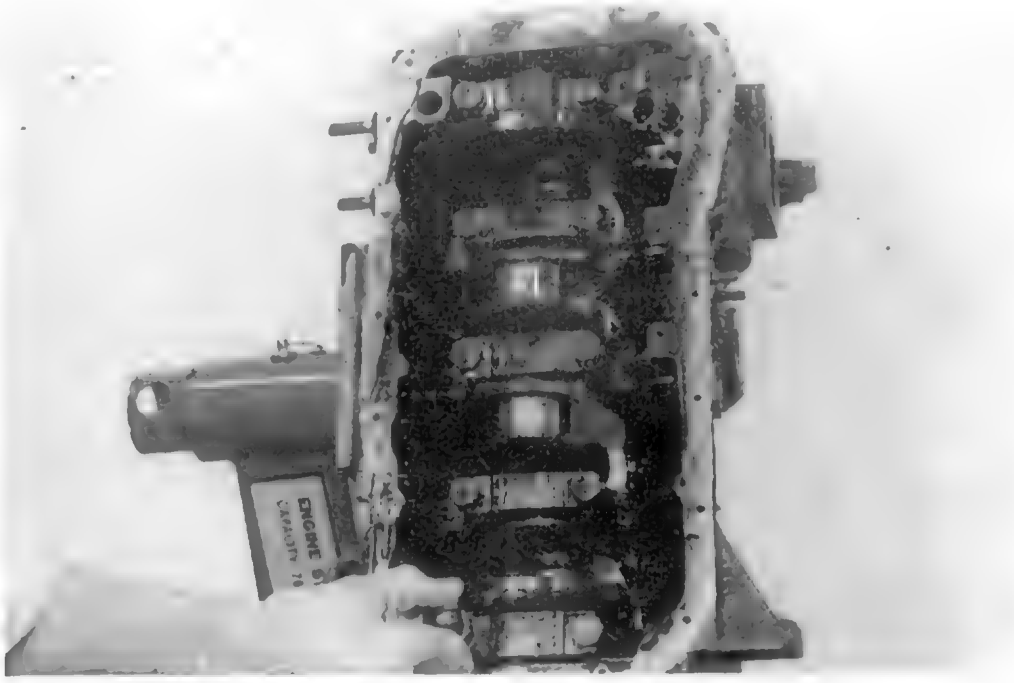
4/23: Loosen and remove con-rod bearing cap nuts. All 1600/1756s require 14mm socket, 2l versions 15mm. Caps are numbered to rods; if stripping two engines, make sure they are kept separate or mark them to identify them to each engine. Rods and caps are not interchangeable. Remove rods in pairs – 1 and 4, 2 and 3. Late models went to 'bolt only' fixing.



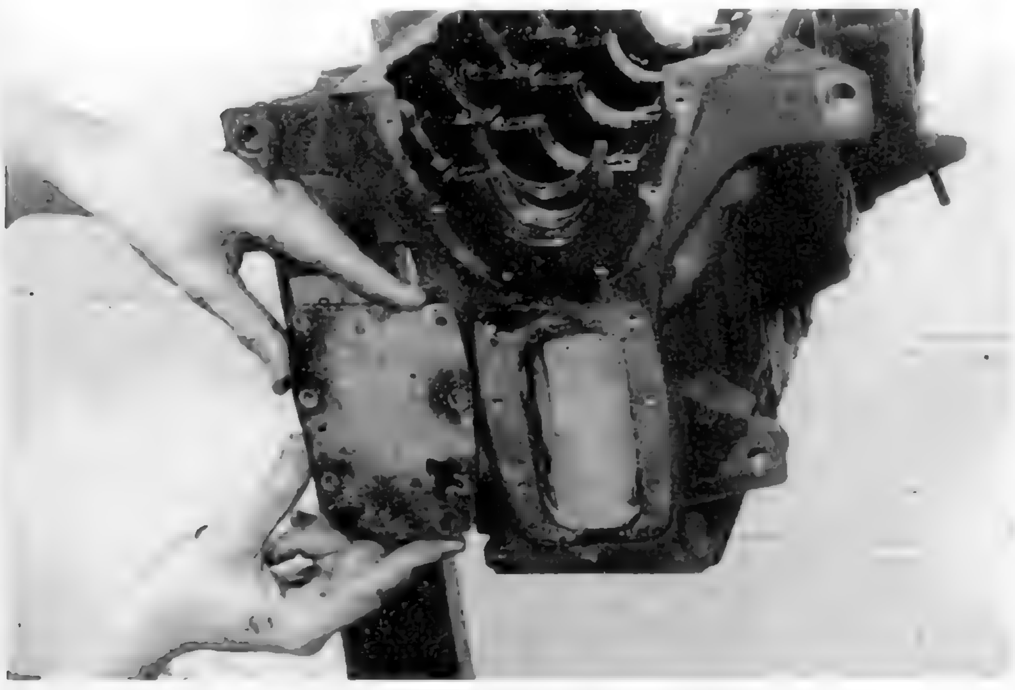
4/24: Turn block through 90° and push pistons and rods out of bores. A short piece of copper tube helps for this. Sharp implements can score rods. Two old flywheel bolts will enable crank to be turned with screwdriver.



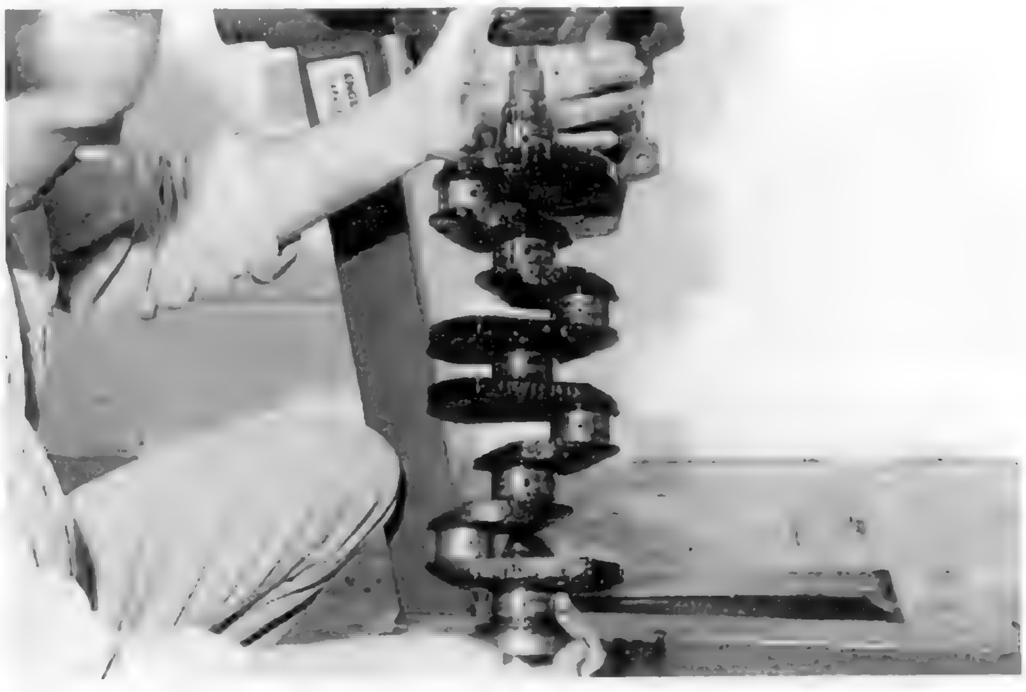
4/25: Catch pistons as they come out of bores. Pistons are flat-top, approx 9:1 CR, used on all 131/132 and Beta (not Volumex) 2l versions.



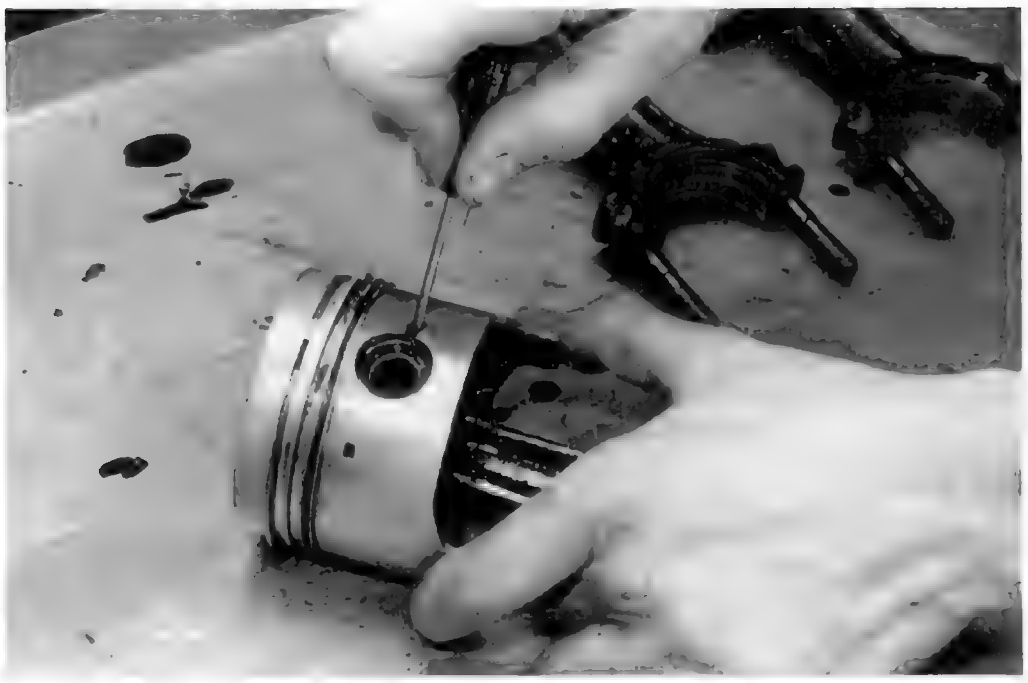
4/26: With pistons removed, main bearing cap bolts can be removed. This 2l version has 9 x 12mm bolts (19mm socket) and 1 x 10mm (16mm socket); 2l has two long bolts on centre cap. Caps are numbered to block and have notches to identify order of fitment; no marking is needed.



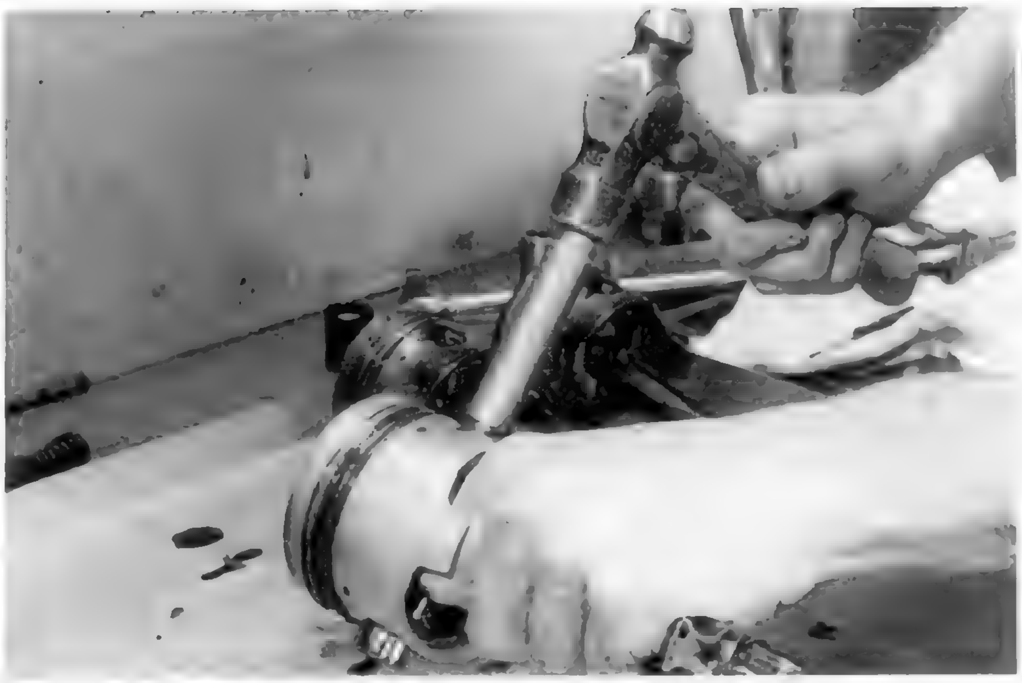
4/27: Remove end plate from block for cleaning and gasket renewal (10mm socket). Note simplicity of mountings for bellhousing – only four bolts required. Inspect main bearings and check condition of housings. If necessary they can be measured with bore gauge, but if bearings are reasonable, don't worry over-much. GCT have never line-bored a TC: if bearings are badly damaged it may be cheaper to obtain another block.



4/28: 2l Beta crank removed. Oily appearance is deceptive – crank was in perfect condition. Keep old bearings and thrust washers. Damage may indicate need for crank grind (possibly retouching of thrust faces) or rod resize. If bearings are being retained, keep them in their housings.



4/29: Wear eye protection or cover clips with cloth when removing. Pistons are size-classed A–E to match bores at factory so if engine is to be honed and re-ringed, keep pistons on rods during cleaning. Scotchbrite is ideal for cleaning old pistons, plus a suitable solvent – Jizer, paraffin, etc. If beadblasting, mask ring grooves, but do not blast with pistons on rods or pin clips fitted – very difficult to clean off blast media.

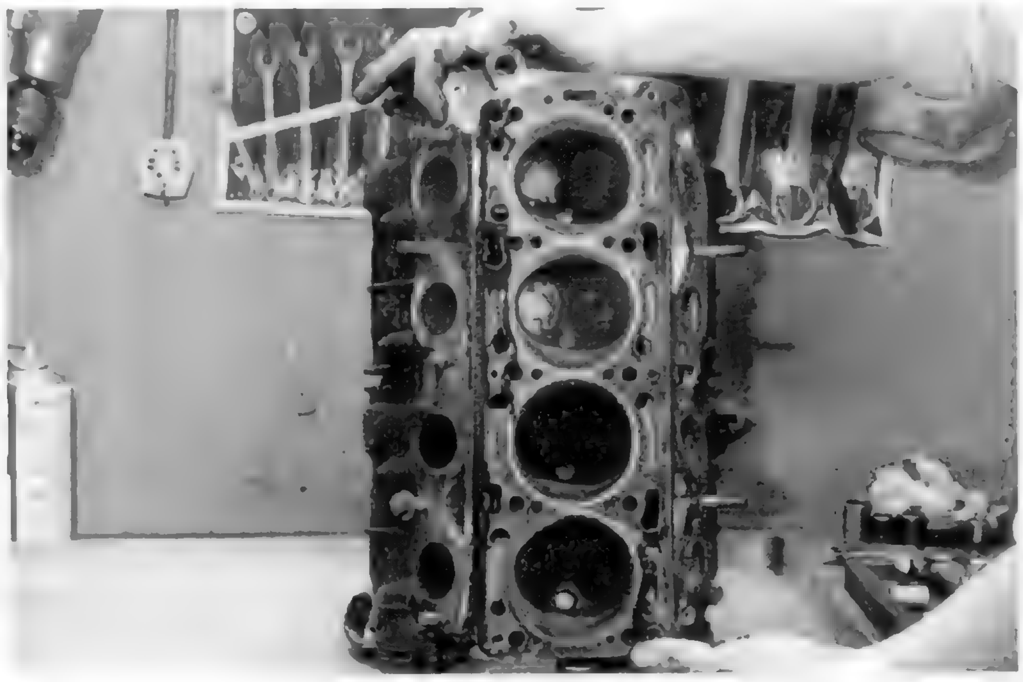


4/30: Use of a suitable size drift allows pistons to be removed from rods. If block is to be rebored, old pistons can be scrapped. New pistons come complete with rings, pins and clips. Place rods to one side if damaged.

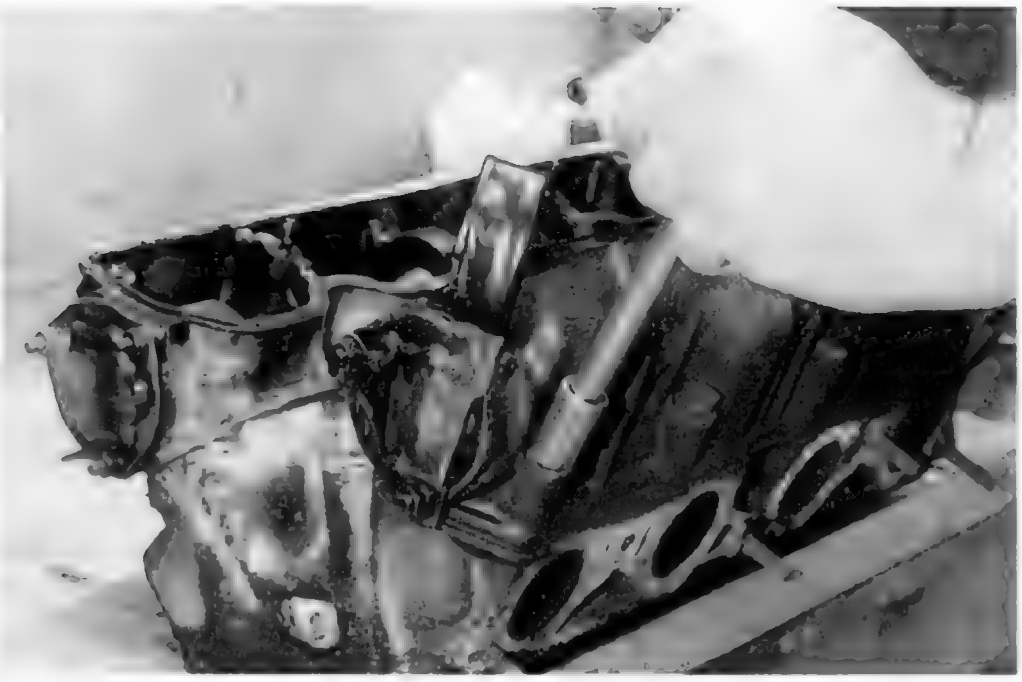
4/31: Tap old con-rod bolts out of rods; always use new bolts and nuts unless they are out of a low-mileage engine. If crank has been fitted with oversize bearings following a regrind, don't take chances with old con-rod nuts and bolts – replace them – they may have been overstressed if engine ran a bearing.

4/32: Remove core plugs – despite their external appearance, which may be good, they can be heavily corroded inside. Tap gently on one side of plug…

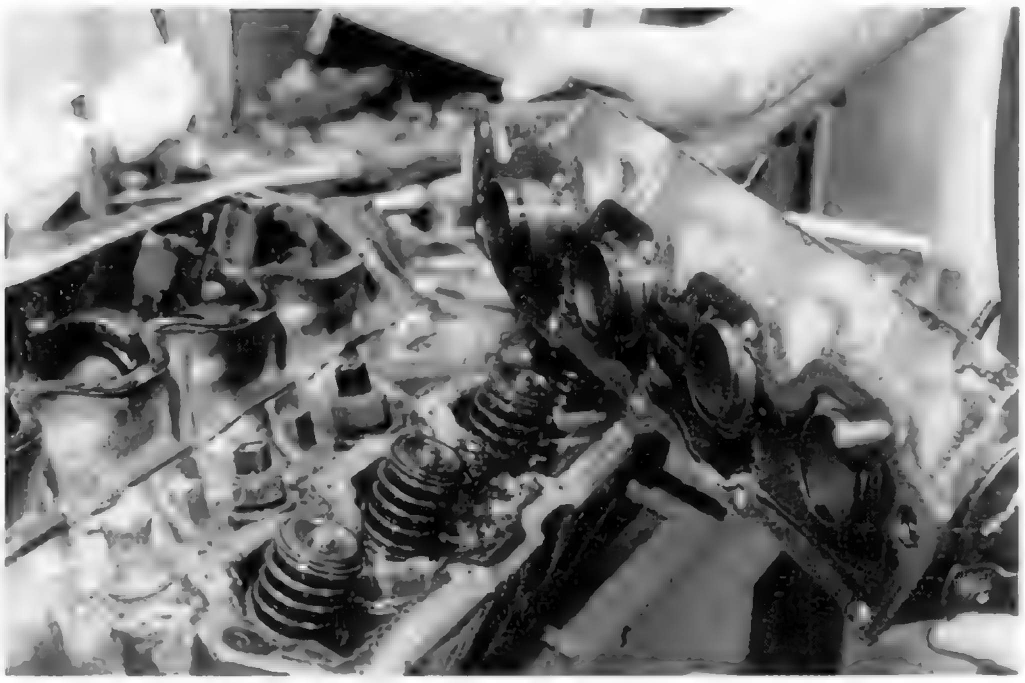
4/33: …to tilt it and then lever out as shown with strong screwdriver. (Author's note: A 'pig of a job' on the 80mm-bore engines!)



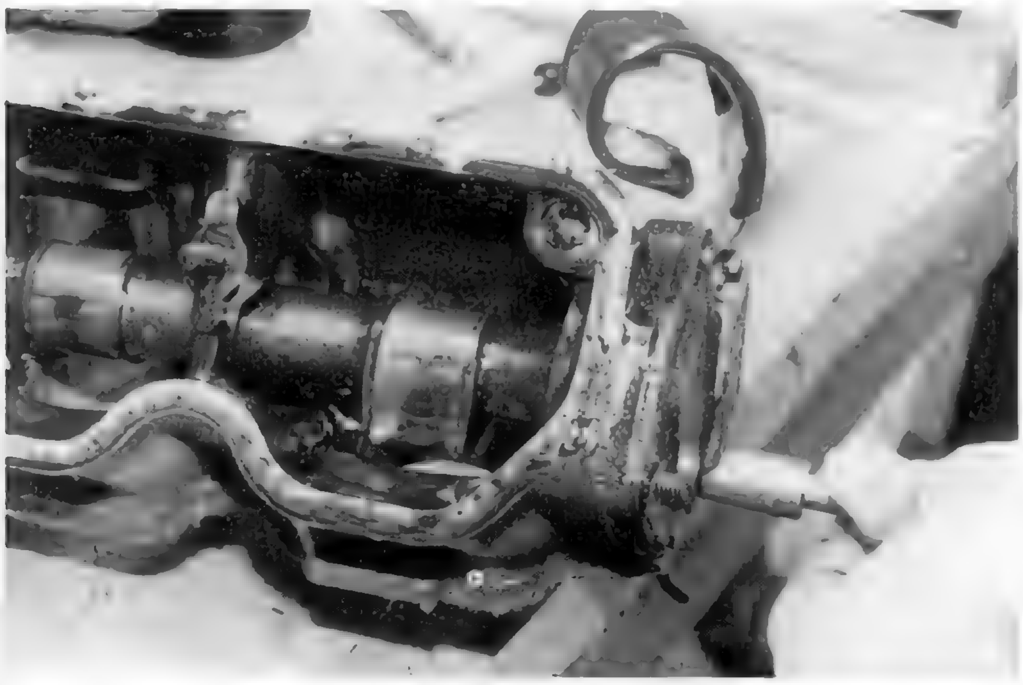
4/34: Cylinder head removed. One of TC's great advantages is similarity of components. All early heads (prior to revised cam layout first used on Delta Turbo 1.6 ie) are interchangeable, although 1608 coolant galleries require modification to align them to 84mm-bore block. Inspect head, block and gasket for damage on mating face.



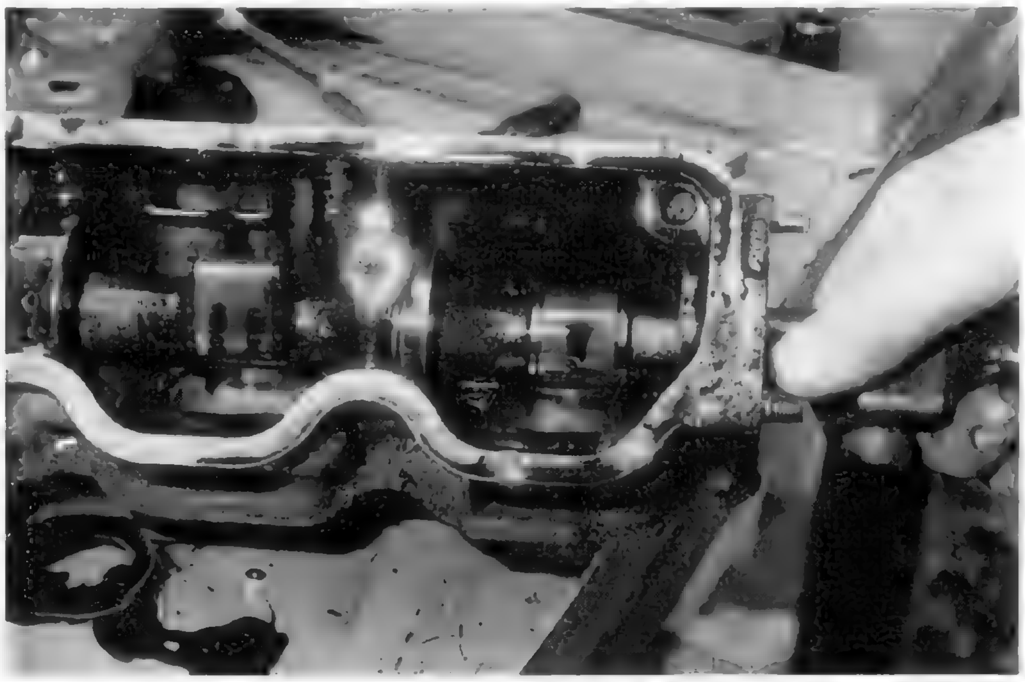
4/35: With head resting on side of tray to avoid bending protruding valves, unbolt cam boxes (13mm socket). Some early models, eg 1608, have studs instead of bolts. Note bolt holes at rear of head which identify transverse unit.



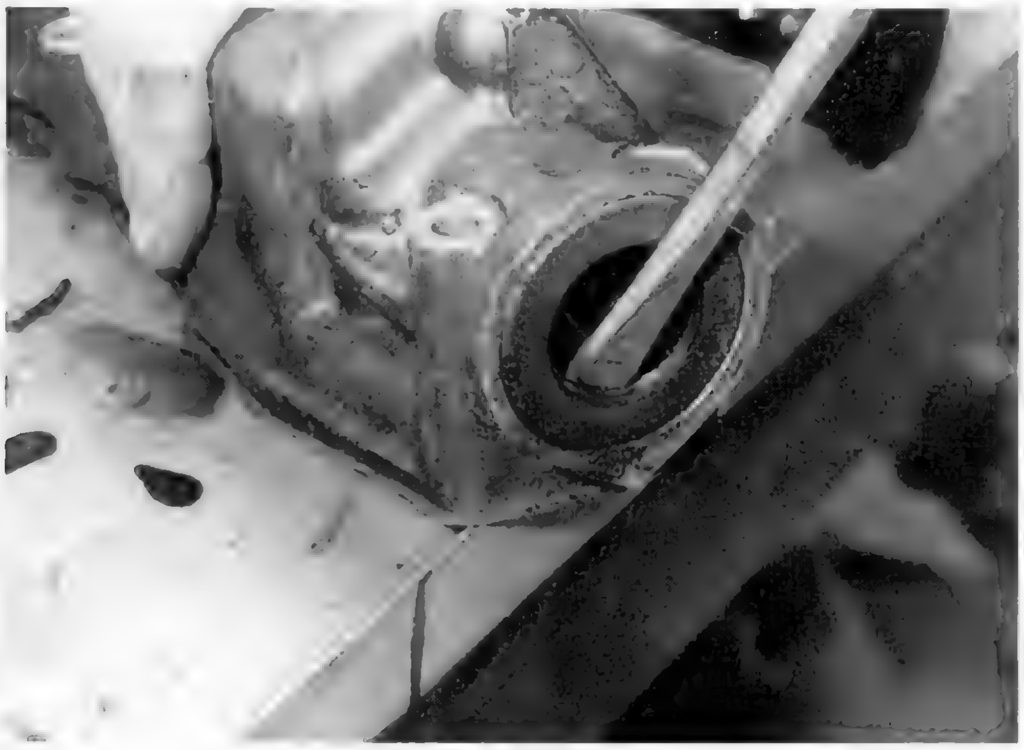
4/36: Remove cam boxes, keeping cam buckets in order, and make sure all nuts and washers are taken out of cam box and accounted for. They can obviously be renewed, but make sure none get lodged in cam box.



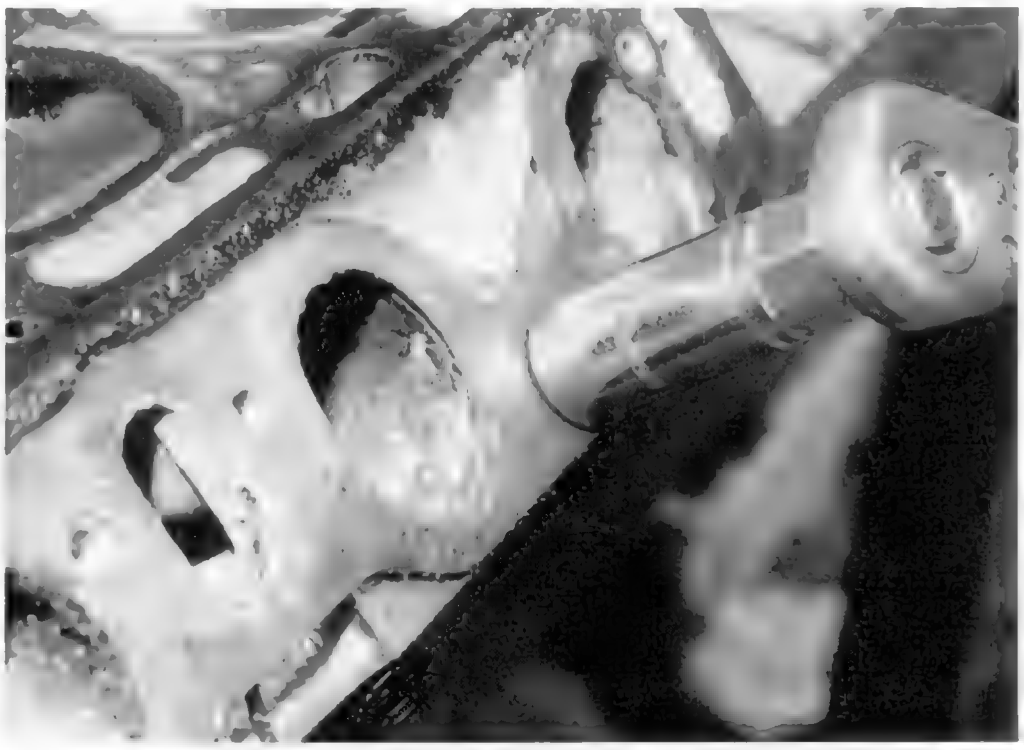
4/37: Undo nuts retaining cam box end cover – some models have end-driven distributor or distributor mounted on top of exhaust cam box (1608/US models) or inlet cam box (Monte Carlo)…



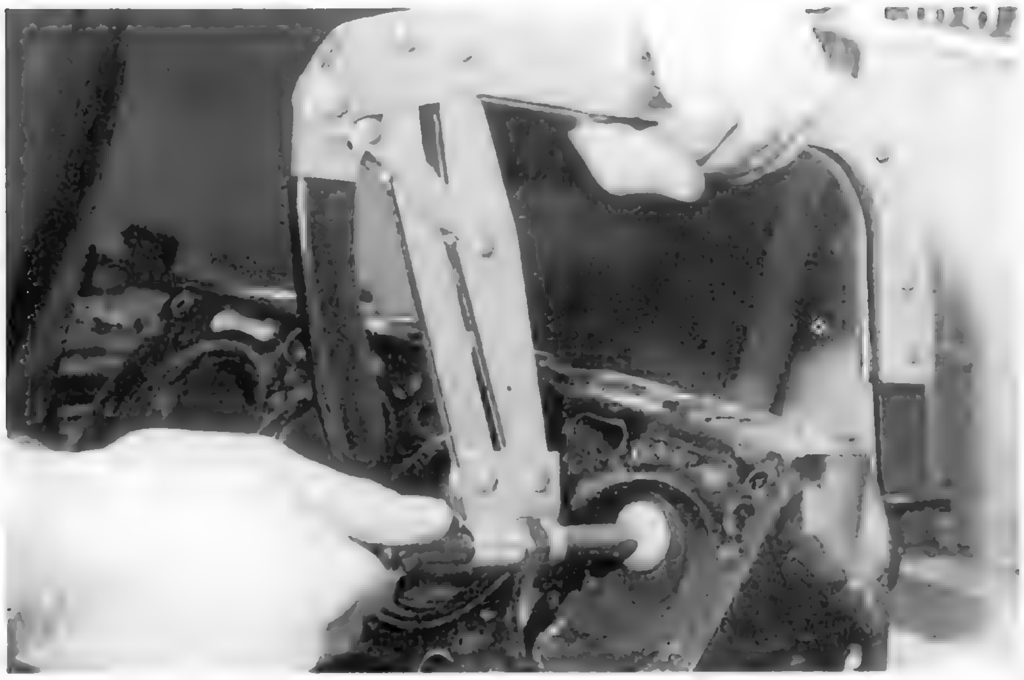
4/38: …and withdraw camshafts. Mark them so they go back in their original housings. Cam buckets and bores can last forever; replacing them is very much a question of cost versus benefit. Cam box camshaft bearing housings go oval after extended use; this and serious scuffing can lead to low oil pressure.



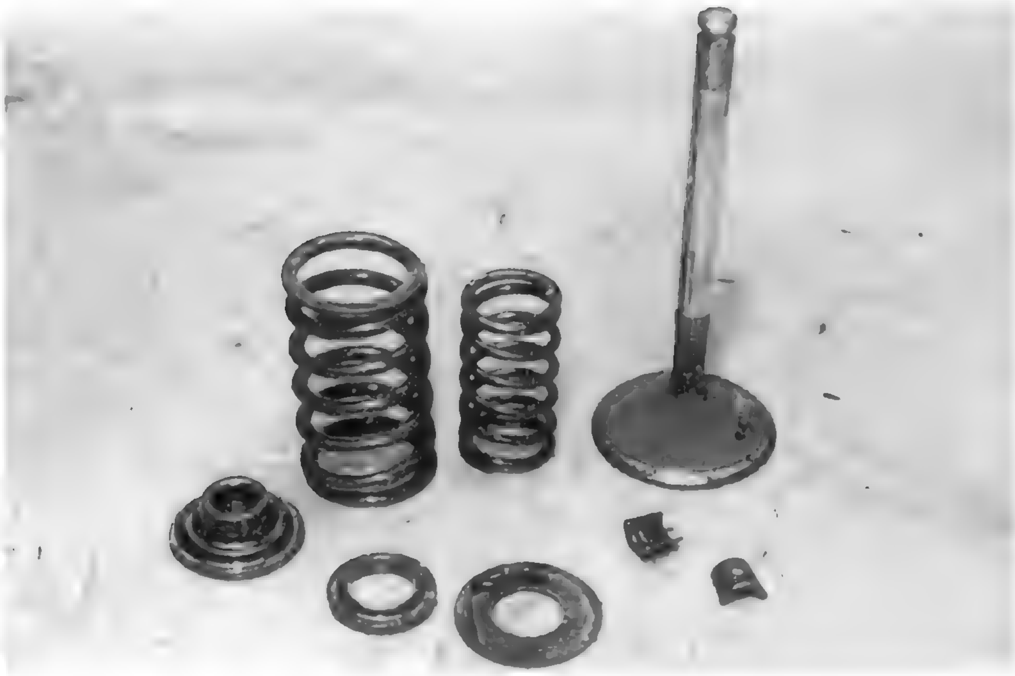
4/39: Lever old cam seal out of cam box with strong screwdriver.



4/40: 'Destud' head. These studs may interfere with head refacing operation, also decarbonizing fluid used for cleaning head may strip off plating. Stud tool is made by Kamasa.

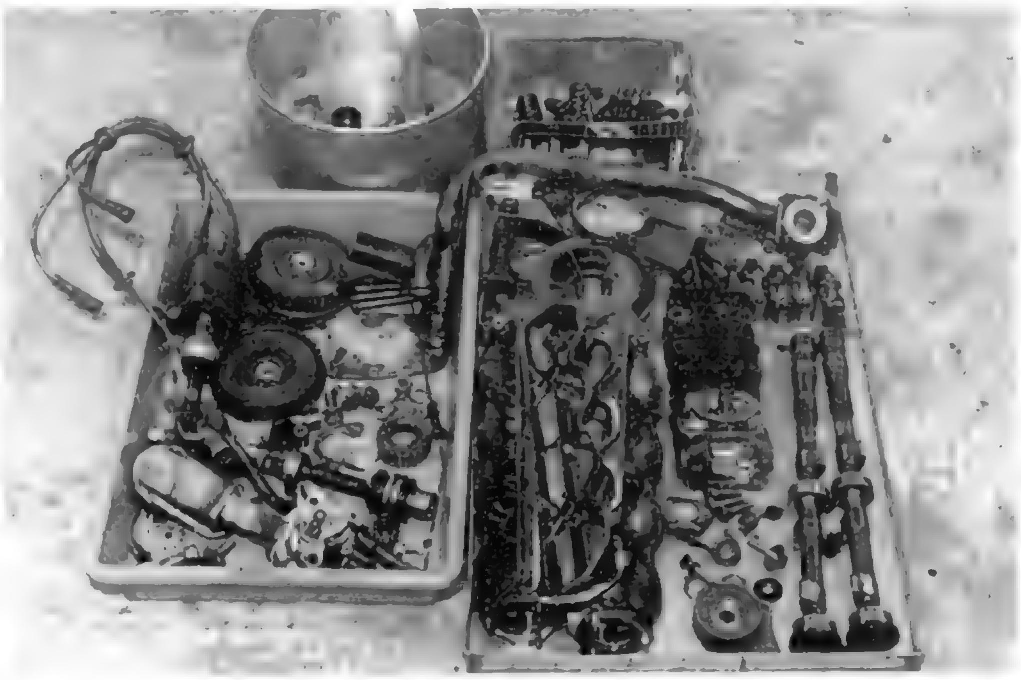


4/41: Compress valve springs and tap tool to free collets. If valves are to be cleaned, refaced and refitted in old guides, make sure they are numbered at this stage.

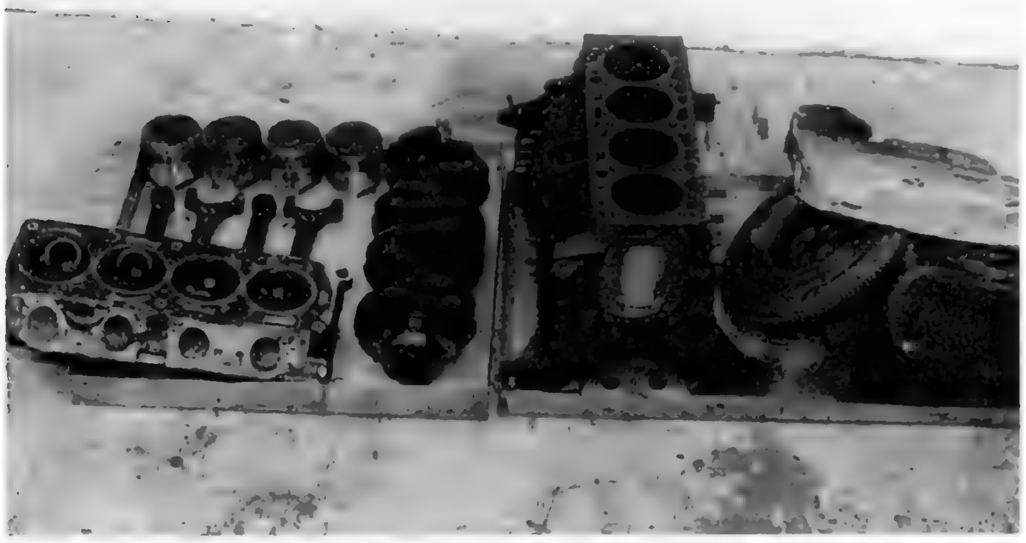


4/42: Valve assembly. Inspect valve stem for wear (with micrometer) and tip and compare with data. Springs can also be tested (see Chapter 6). This inlet valve is relatively clean – heavy fouling indicates worn guides or perished stem seals. Collets, caps and spring seats rarely need renewal.

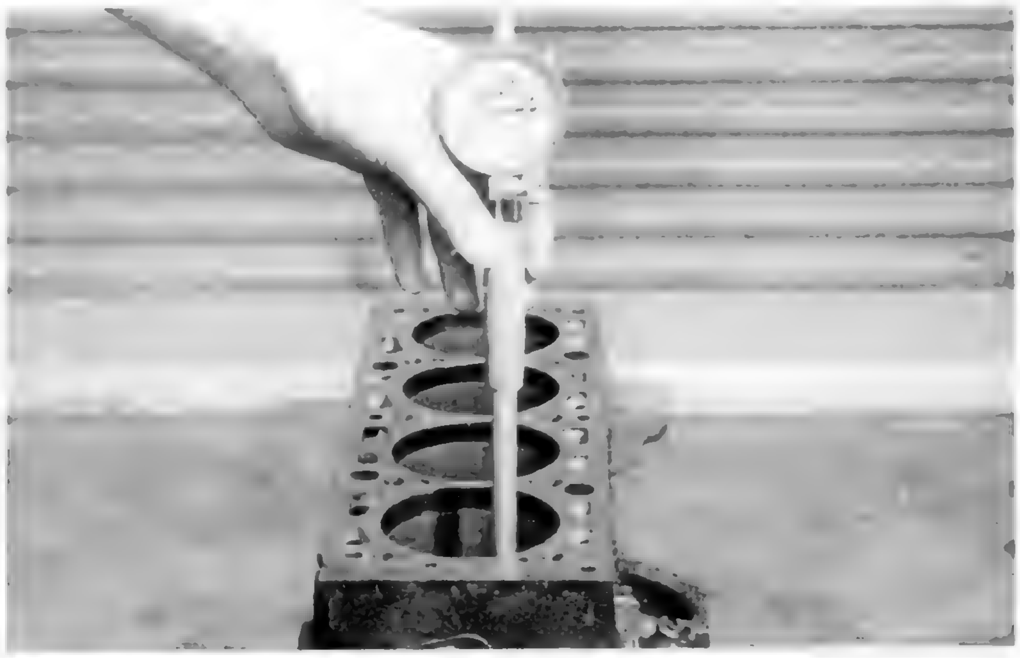
4/43: Carefully remove shims with slim screwdriver. If valve seats are not recut, original skim thicknesses can probably be re-used. Shims are available from 3.20mm–5mm normally, but thicker Volvo shims are available, though extra weight is undesirable.



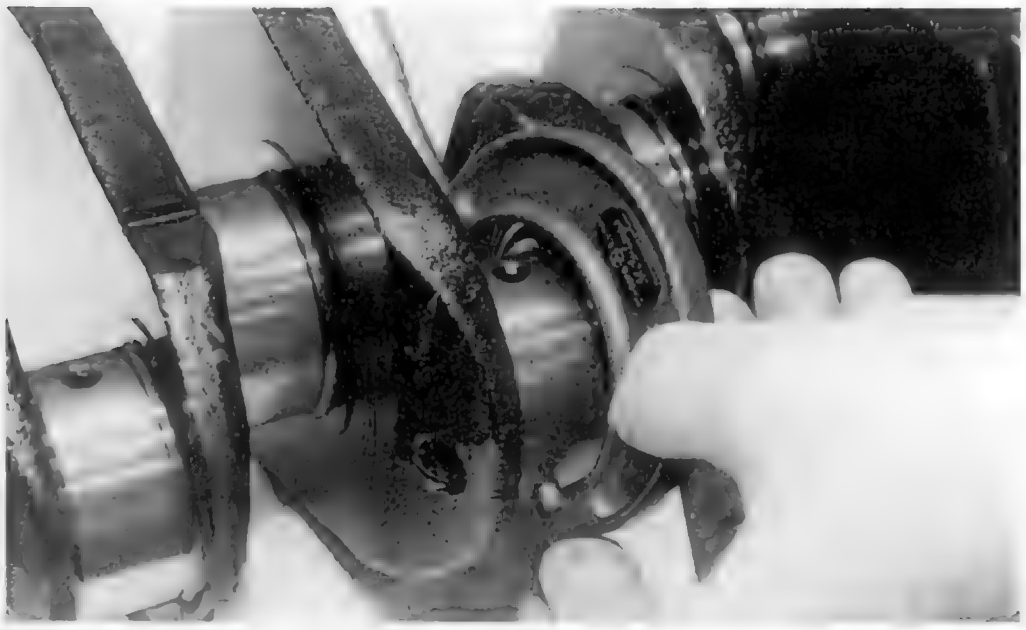
4/44: Fully stripped engine can now be sorted into groups for cleaning and inspection...



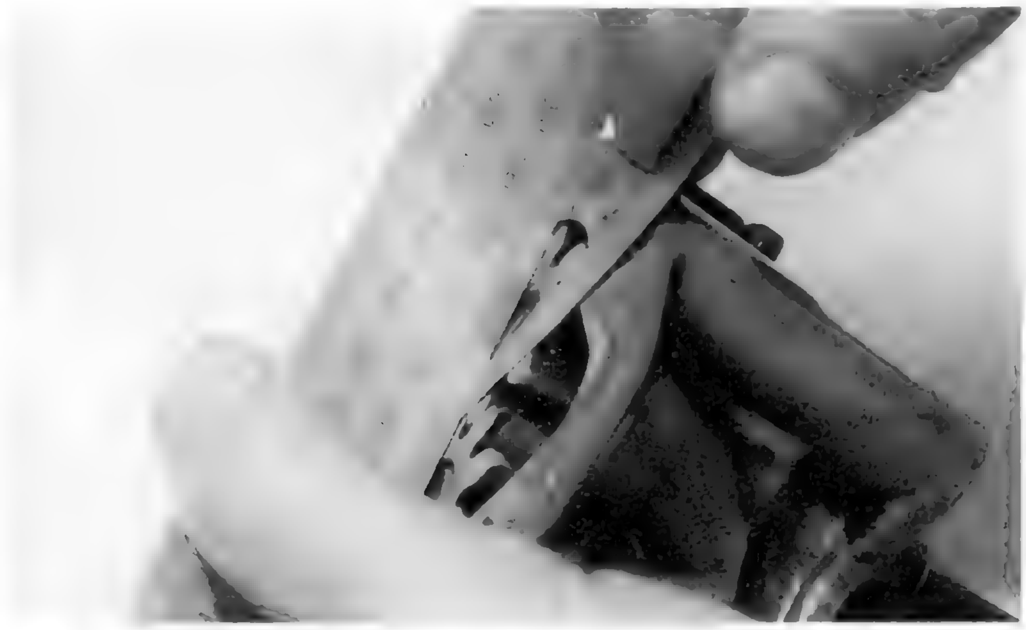
4/45: Plated parts can be cleaned in solvent and replated (bright zinc and colour passivate to give attractive 'rainbow gold' colour). Heavily carboned parts, eg rods, main bearing caps, cam boxes, crank, need to be decarbonized overnight or soaked in cleaning fluid prior to rinsing off. Distributor, tensioner must be carefully cleaned to avoid damaging bearings. Certain solvents damage plastics; take care with crank end bearing if it is not to be renewed.



4/46: Checking bores with bore gauge. Gauge in this case is set at 84.00, needle has moved anticlockwise from 0 indicating block is worn to 84.04. Worn bores have wear ridge at top (below carbon deposits) caused by rings. Removing this ridge is tricky and stepped top rings are hard to get – a rebore is usual answer. Smallest reading is true diameter.



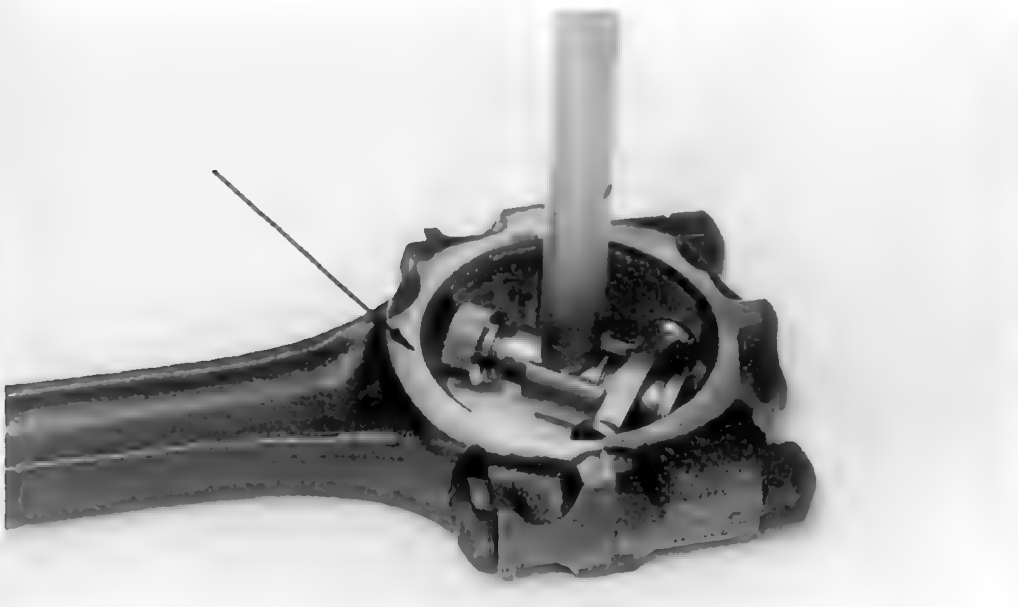
4/47: Micrometer is essential for measuring crank. Note sizes of all journals with gauge as shown and also at 90° to it. Watch out for odd factory undersize journals – normally stamped on a flat on crank web (left of picture). Bearings are not available for these so regrind will be needed if old bearings are not being re-used (not a desirable practice!). A good crank may only require a light polish. If thrust washers are badly damaged, inspect crank thrust face – grinding may be needed.



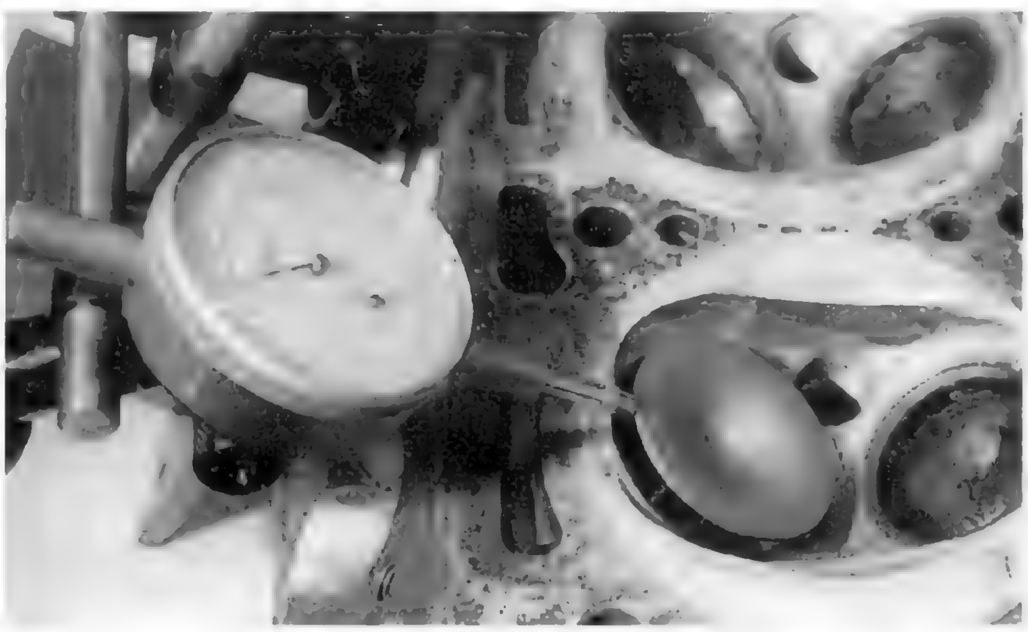
4/48: Use straight edge and feeler gauge to measure end float of oil pump gears and compare with data. Check casing for scoring, which will not show up with measurement. This is usually caused by damaged bearing parts going through pump. Also check condition of oil pump spline – if worn, replace – they can shear.

4/49: Measure rotor-casing clearance. Do not worry too much about backlash between gears – new pumps sometimes have more than they should. If any of these checks show excessive wear, discard pump or risk low oil pressure (or none at all if shaft breaks)! Checking procedure is same for nose-driven pumps.

4/50: Purpose of measuring piston skirt is to identify skirt-bore clearances. Cast pistons usually have 1½–2thou", forged may be around 2thou" for low-expansion alloys, 4–6thou" for high-expansion types. A 4thou" skirt-bore clearance is really the absolute limit for cast pistons, and remember that big bores will give high ring end gaps unless oversize rings are used (and gapped to fit).

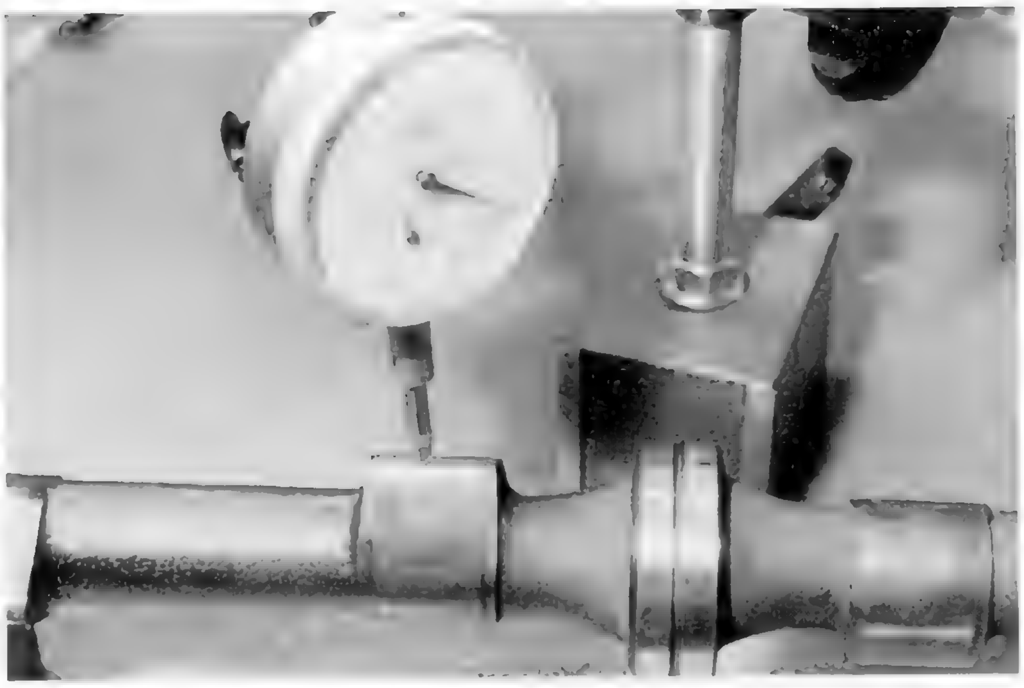


4/51: Bore gauge or digital vernier can be used to measure big-end bores, which go oval after extensive use – leading to low oil pressure. Rods can be honed back to size without detriment to strength, but no con-rod has indefinite fatigue life. Check small-end bush: a well-polished appearance is usually a good sign – use piston pin to check running clearance. Also inspect rod 'cheek' (arrowed) – light damage can be dressed with 220 grit.



4/52: Amount of wear in stem and guide can be approximately determined by holding valve open 5mm and measuring free play with dial gauge. This set-up had 0.35mm, which is about limit before new guides should be considered. Best way to check valve for true is to put it in a valve refacing machine. Combustion chamber is early 131 type.

4/53: Measuring cam journals with micrometer (see data). Cam wear is not a problem with TCs provided proper oil changes (and levels!) are maintained. Bearings are fed from oil galleries in head/cam box, lobes run in oil bath. Production cams are high-grade cast iron.



4/54: Cam lift can be determined by placing cam journals in V-blocks and measuring lift from base circle to tip of cam lobe: deduct valve clearance to obtain true lift. Out-of-spec of ±0.2mm is quite acceptable for engine using standard cams; more important is condition of lobe. Cams with no running clearance will have worn base circle too.

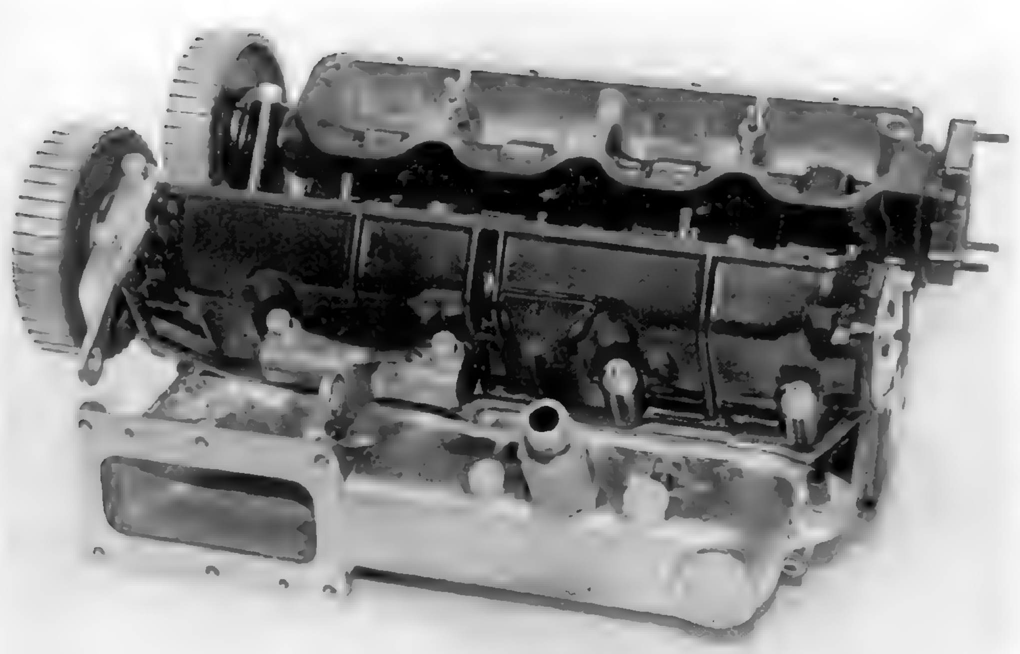
4/55: Measuring step on flywheel with straight-edge and feeler gauge. Step height needs to be 20thou" (±2), but condition of friction face equally important – a pitted, corroded face will damage clutch. (Same technique can be used for assessing flatness of cylinder head – 3thou" is absolute limit.) Check condition of ring gear at this time (they can be renewed) and inspect area around crank bolts for cracks. Surface cracks on friction face may come out with regrinding operation, small heat cracks (up to approx 5mm long) are not critical on a low-powered engine, but on one being used at 7000–7200rpm (the realistic limit for a cast iron item) replace flywheel.

4/56: Chronic wear inside bucket. This can cause valve to tilt during operation; bucket must be replaced. Wear is usually caused by overly hard valve springs, gross petrol contamination of oil, or chronic oil neglect/worn guides.

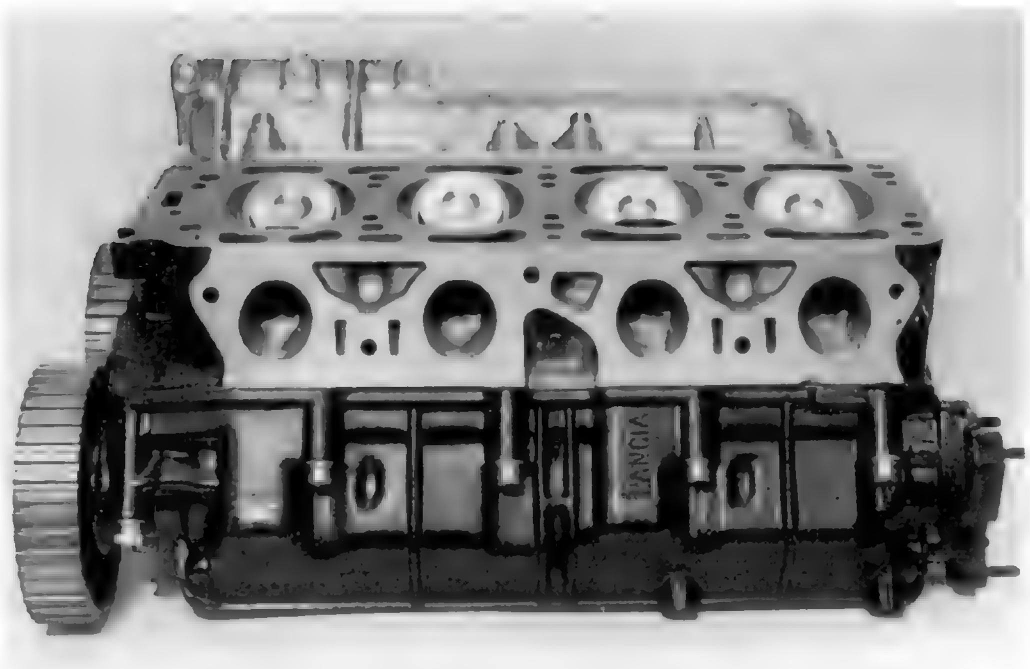
CHAPTER 5

CYLINDER HEAD PREPARATION

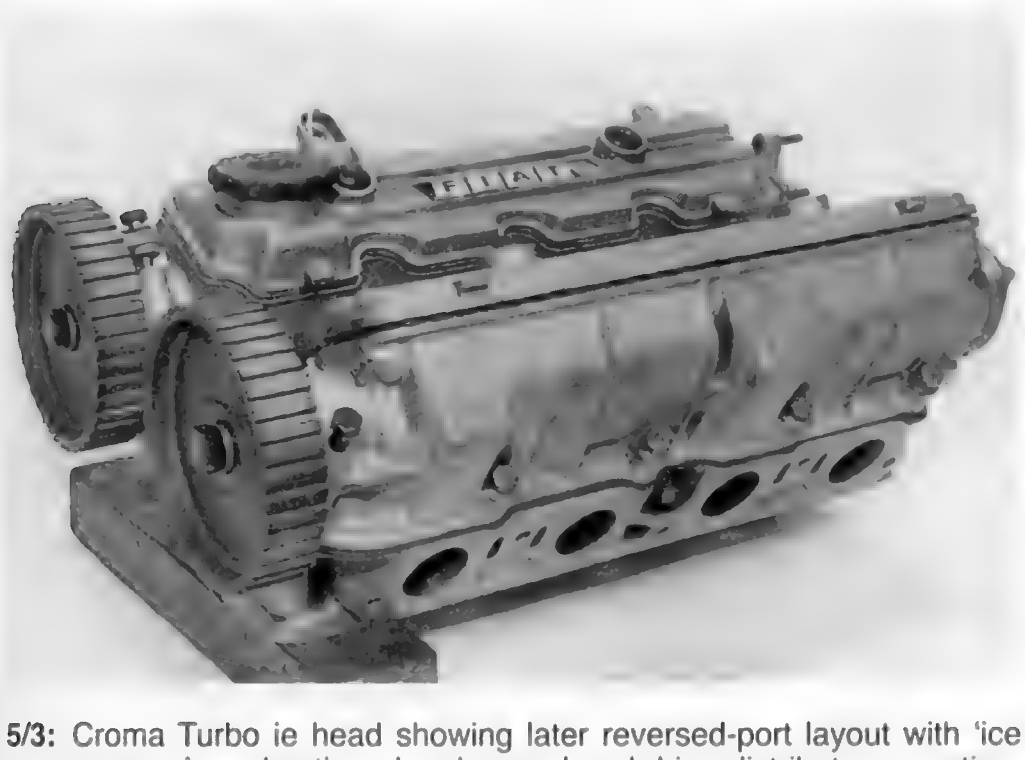
Analysis, modification and building up



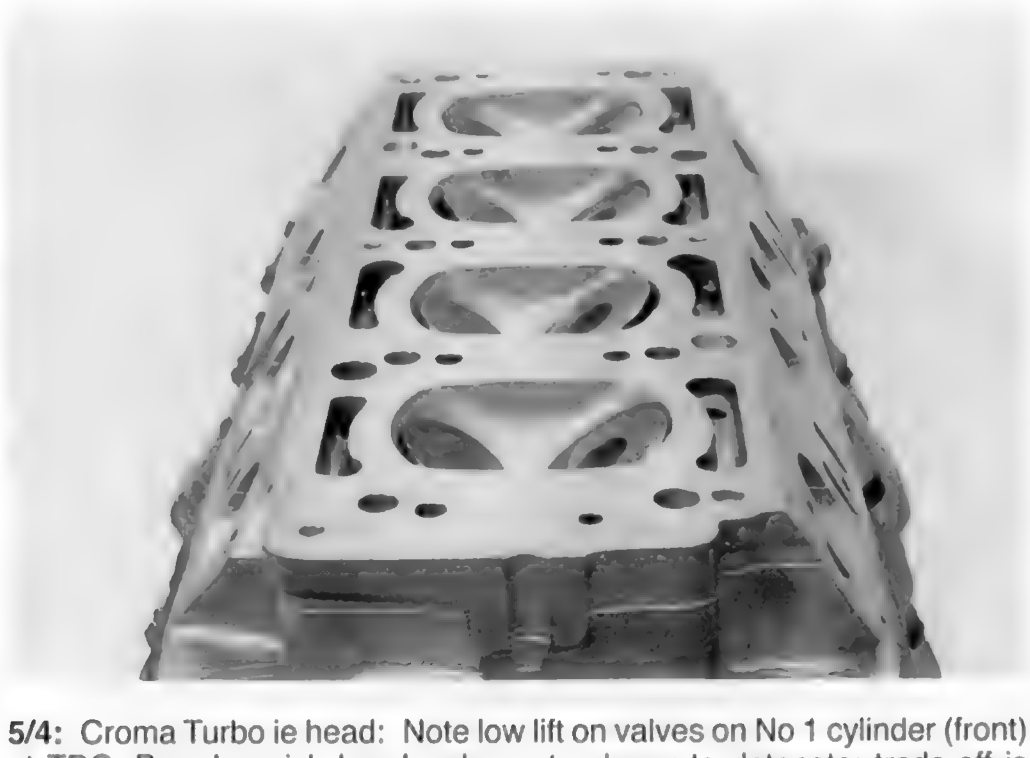
5/1: Lancia Volumex cylinder head. Supercharger bolts to inlet manifold, relief valve is set at 15lbf/in^2 – blower only produces around 6. Volumex has end-drive distributor, 30mm flangeless cam wheels. Note use of stainless steel 'cap heads' to replace standard hexagonal bolts. Inlet manifold is prone to warp and should be bonded to head with gasket and silicon sealant; also vital that when blower is fitted, mountings are secured in proper sequence.



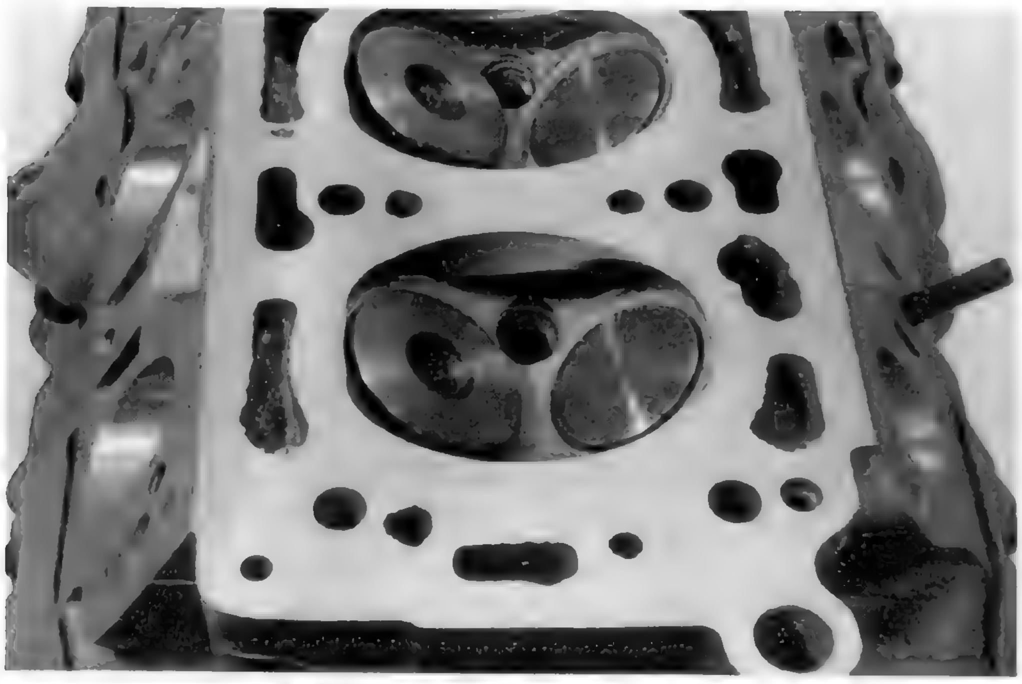
5/2: Inverted shot of Volumex head shows late-type combustion chamber, similar to late 2l Beta, but with larger volume. This model has 44 race inlet valves, production 36mm sodium-cooled exhausts, fully blueprinted seats and is ported. Note low lift of valves on No 1 cylinder (cams are set at No 1 TDC).



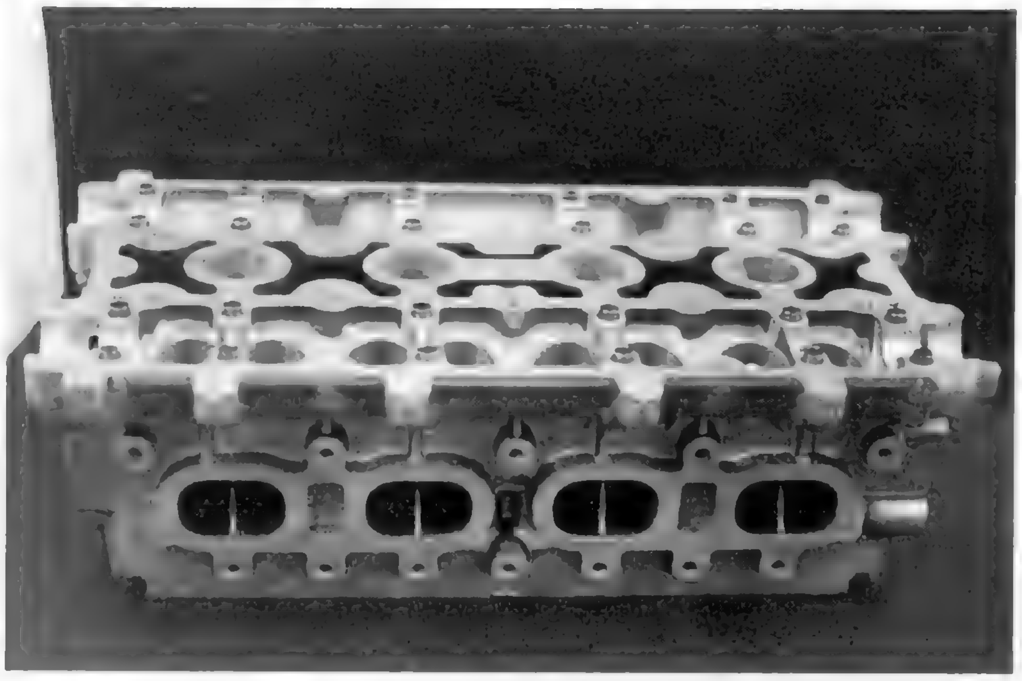
5/3: Croma Turbo ie head showing later reversed-port layout with 'ice cream scoop' combustion chamber and end-drive distributor mounting. Camboxes and covers were redesigned for these heads and are not interchangeable with early models. Weak 6mm dia studs retaining covers were retained – don't overtighten.



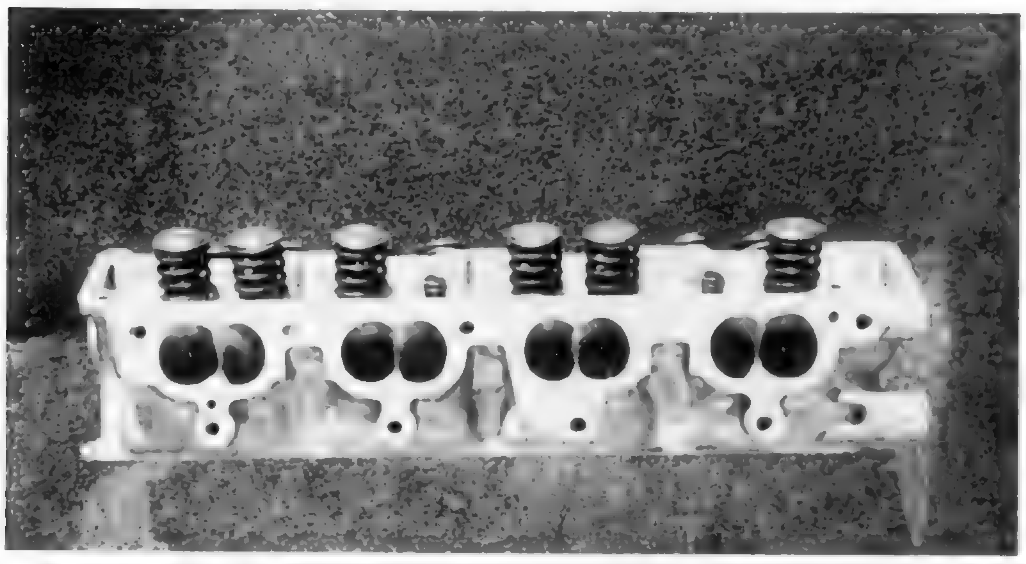
5/4: Croma Turbo ie head: Note low lift on valves on No 1 cylinder (front) at TDC. Broad squish band reduces tendency to detonate; trade-off is reduced airflow due to shrouded valves, but being a turbo, fuel/air is forced in under pressure. Valve sizes are 43.5 inlet/36 exhaust (sodium-cooled); 8v turbos all had bronze exhaust guides (cast iron inlets).



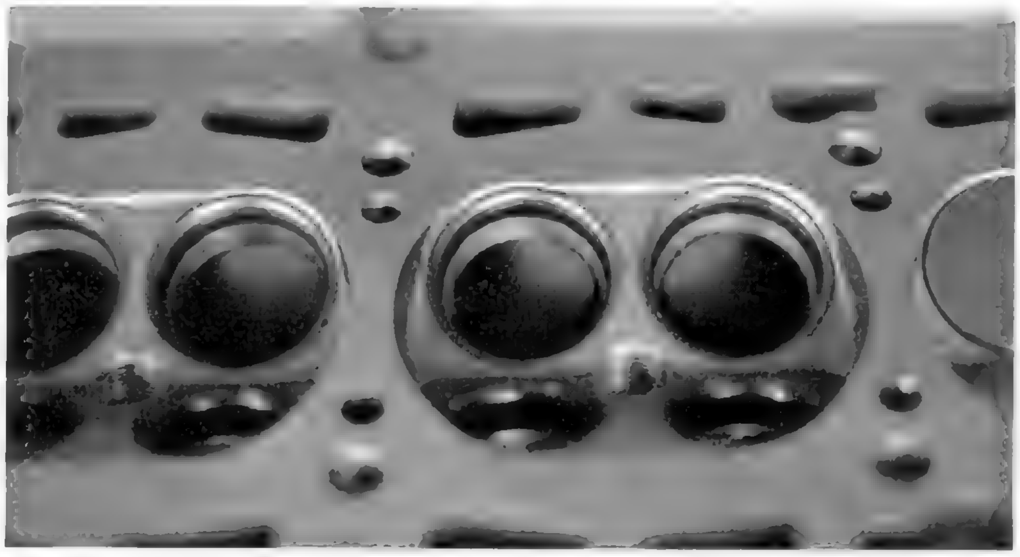
5/5: Head used on late 131 2l model. Also found on Beta 2l (but with larger inlet ports). All these heads used 41.8 inlet/36 exhaust valves, cast iron guides. Corrosion around coolant galleries caused by lack of anti-freeze in circuit. Note dowel locating recess in front RH oil gallery. Head locating dowels (one also at rear LH corner) must be fitted or head gasket will shuffle about and blow. Refacing operation on head has taken cutter through inlet valve seat insert. With a tungsten carbide cutter, this is no problem. Alloy heads can also be satisfactorily wet-ground. Remember, however, that refacing head (or block) reduces valve-piston clearances.



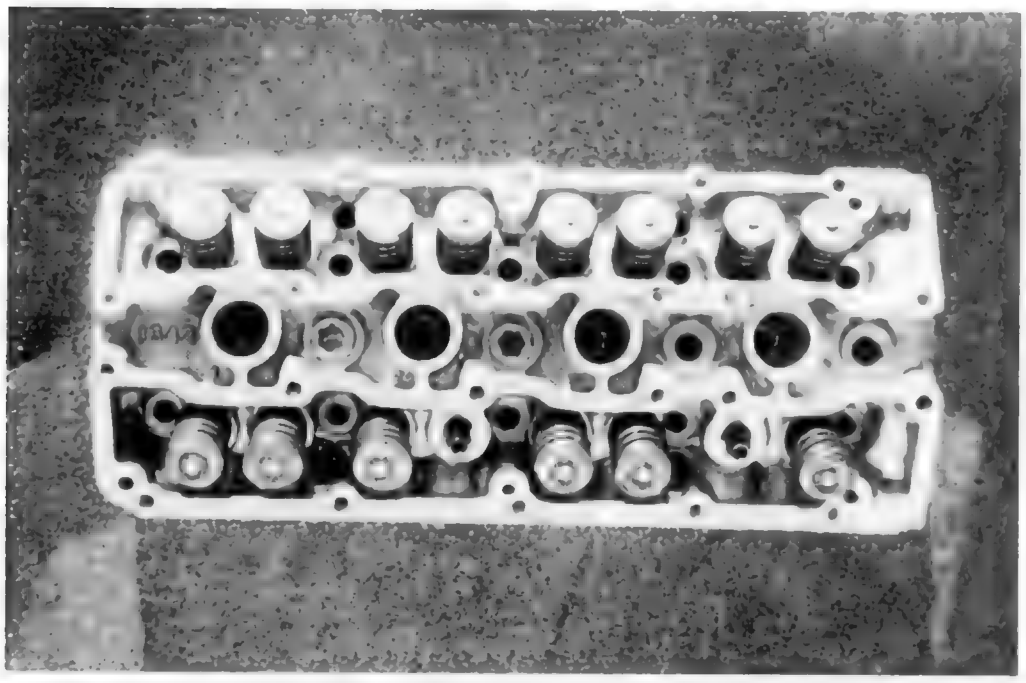
5/6: Late 16v head (ex-Thema/Integrale) features bronze guides, nickel bronze valve seat inserts, 21-4n chrome stem valves (sodium-cooled exhaust) with 7mm stems. This head fits early block, although 16v-type pistons are needed.



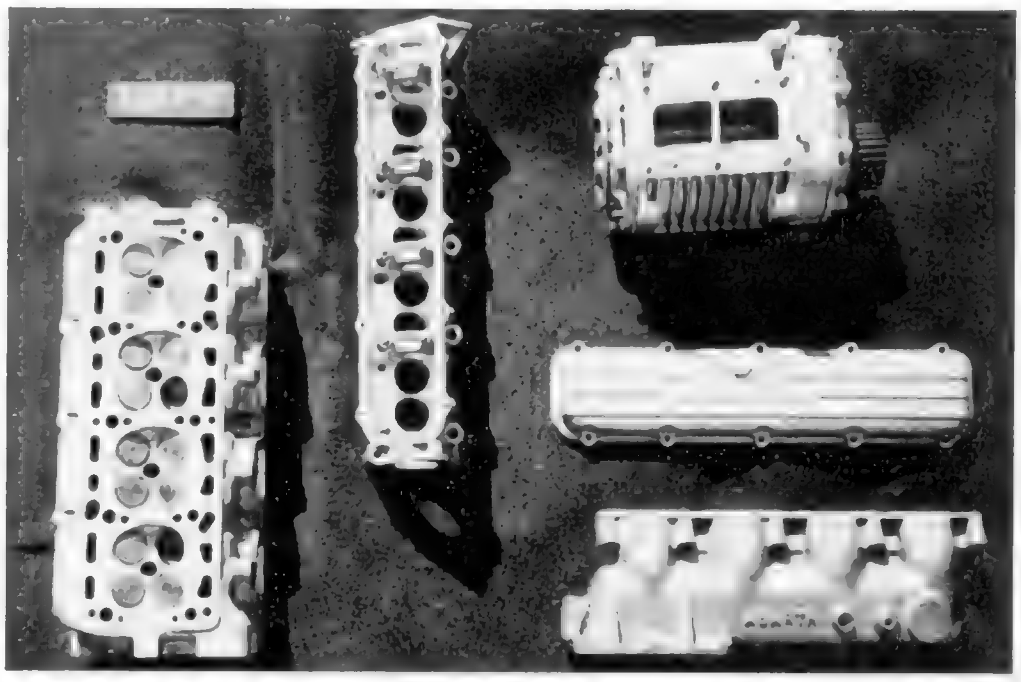
5/7: Early 16v head. Note different design of inlet ports and use of removable cam carriers. These heads are now extremely rare and sought after. (Photo Tom McGaffigan)



5/8: Close-up of late 16v head shows similarity of combustion chamber to early design. Location of plug in centre of chamber allows even expansion of flame front in all directions. Major benefit to be had from relieving 'short side radius' in both inlet (shown) and exhaust ports. Incoming mixture breaks away from port wall – resulting turbulence reduces cfm.



5/9: Early 16v head. Fiat casting technology has been equalled (eg by Cosworth), but never surpassed. Later head is taller due to integral cam carriers – designed to simplify manufacture with modern CNC techniques. (Photo Tom McGaffigan)



5/10: Beautifully executed components of 16v Lancia 037 head. (Blower is Volumex, not 037.) (Photo Tom McGaffigan)

KEY DATA – 8V CYLINDER HEAD TYPES					
CASTING NO	VALVE SIZE (mm) IN	EX	INLET PORT DIA (mm)	USED ON	COMBUSTION CHAMBER TYPE/ NOTES
4304781	41.8	36	34–32	1592 (124 SPORT 132 GLS)	LATE
4268803	42.4	36	36//	1608 124 SPORT	EARLY – NEED COOLANT GALLERY MODS TO FIT 84mm BORE ENGINES
4326319	42.4	36	32–30	AS ABOVE	AS ABOVE
4372297	41.8	36	32–30	1585 BETA, 1585 2*l* 131	LATE
5992129	41.8	36	34–32	1585 131, DELTA 1.6 HF TURBO (carb)	LATE – ALSO SEEN ON NON-UNLEADED 2*l* US SPIDER
4462603 4314402	43.5 42.4	36 36	36–33 36–35	105 TC (1585) 1756 (124 SPORT) EARLY 131 2*l*	LATE – BIG-VALVE TYPE LATE
4371507	41.8	36	VARIES FROM 35–32 TO 32–30	131 1585/2*l* 132 1756/2*l* MONTE CARLO	CAN BE EARLY OR LATE
4372748	41.8	36	35–32	MONTE CARLO, 131 2*l*	LATE
4325215	41.8	36	35–33	AS ABOVE	EARLY
4372291	41.8	36	35–33	124 SPIDER 2*l*	UNLEADED US VERSION (LATE)
5936188	41.8	36	36//	VOLUMEX VERSIONS	LATE – SODIUM-COOLED EXHAUST VALVES
4372281	41.8	36	36–35	BETA 2*l*, SOME MONTE CARLO	LATE
5992303	43.5	36	36–34	130 TC, 105 TC, DELTA/ PRISMA 1600 GT	LATE – 130 TC HAS LARGER PORTS

This list is not exhaustive, but it covers the main types seen at GCT over the years.

The majority of the heads have inlet port taper, indicated, for example, by 35–33, the diameter of the port at the venturi near the valve guide being smaller than the port face. All models have an exhaust port diameter of approximately 31mm. Some early models, *eg* early 131, 1608, have a valve guide diameter of nominally 15mm rather than 14mm. All models have 8mm diameter (nominal) valve stems, the same valve caps, springs, collets, caps and cam buckets. (Models with 15mm guides have spring seats with a larger ID than later types.)

All the 8v late 'reversed port design' heads have the distinctive 'ice cream scoop' (author's description!) combustion chambers, broad squish bands and 41.8/36 valves.

Port dimensions are inlet 34–32mm, exhaust 31mm. These heads can be used on the early (*eg* 131) type block if pistons with the appropriate valve cutouts are used, but do require the use of the OE cam boxes.

On these models the coolant outlet is at the back of the head (adjacent to No 4 cylinder). Obviously the head gasket must have correct orientation of the coolant passages.

These heads are to be found on the Delta Turbo *ie* (1600), Croma Turbo *ie*, Tempra 2*l ie*, HF 4x4 (Turbo) and 8v Integrale. Their combustion chambers do not flow as well as their earlier (*eg* 130 TC) n/a counterparts, but they can be opened out, particularly when being used in a normally aspirated conversion where the revised port layout gives better access for the inlet or exhaust system.

Turbo versions of the late heads have sodium-cooled valves and bronze exhaust guides. Exhaust and inlet mounting points are the same as on the earlier heads.

The valve sizes on the later 16v heads are 34.5mm (inlet), 28.5mm (exhaust).

Valve inclination

(The angle quoted is between the valve *head* and horizontal, *ie* head face.)

8v (early and late):	inlet	31.5deg
	exhaust	33.3deg
16v (late)	inlet	23.5deg
	exhaust	22.0deg

[*Author's note:* GCT have only rebuilt one 124 Sport 1608 (BC) engine – not surprising since they are quite rare. This particular model had stainless valves (not chromed) and compatible bronze guides. I would be interested to know if these were standard production fitment.]

CYLINDER HEAD RECONDITIONING

Reconditioning the head

Basic reconditioning should include:
- Cleaning
- Valve reface
- New guides (if required)
- Seat recut
- Head minimum reface

Cleaning and inspecting

The best way to clean the 8v TC head (plus cam boxes etc) is by beadblasting with suitable media, *eg* Guysons Honite. This medium consists of spherical granules and 'peens' the surface of the alloy without damaging it. It easily copes with carbon deposits and leaves the head clean and easy to work on. It has the added advantage that the process closes up the pores of the metal surface, making it less absorbent to oily fingerprints!

Prior to beadblasting, the head and valves should have all traces of oil removed and all bolt threads should be plugged with old bolts. Removing oil and soft carbon can most easily be done with an overnight soak in decarbonizing fluid followed by washing with a water-soluble degreaser. Water wash and air dry follows. Some degreasers (*eg* Comma Hyperclean) contain effective anti-corrosion additives,

but if the valve guides and seats are being retained it helps to spray them with WD40.

Beadblasting follows, but first scrape off old gasket material (early sets contained asbestos, which is toxic). Thorough beadblasting will easily reveal any surface cracks. This is very rare on TC heads, but if it happens it will usually be in the area of the coolant galleries on the exhaust side. GCT have never needed to resort to pressure testing on heads (or blocks). Don't beadblast 16v heads: the structure is very hard to clean and beadblasting media is almost indestructible. It will rip bearings and oil pumps to shreds if it gets into the oil circuit. The dust generated by the process tends to absorb water and form a hard compound, so wash the head after blasting. The latest vacuum blast cabinets use a mixture of hot water and beads to achieve the same purpose and are very effective.

When a TC head is cleaned with water-soluble solvent and then water-washed and air-dried, cracks will show up quite readily since the solvent is retained in the cracks and shows up as a stain on the metal surface (dirty solvent is more effective than clean!). This is a very easy way to spot cracks in ports and other 'hard to see' places.

Check the condition of the plug threads and combustion chambers. Damaged threads (6mm, 8mm and plug threads) can be repaired with Wurth's Time Sert system **(5/11)**, which employs an insert threaded on the ID and OD, which locks into place. This is superior to Helicoil (which, it must be said, was used with success for many years) in that the top thread cannot come loose.

Check the condition of the head face using a straight edge (*eg* steel rule in good condition) and feeler gauge. The maximum out-of-true allowable is 3thou". Check along the head and across it. If it exceeds 3thou" or is scored where the fire ring of the head gasket sits, it must be refaced.

Do not worry overmuch about minor detonation damage in the combustion chamber unless it badly undercuts the valve seat inserts. In this case, welding and new inserts may be needed to stop them falling out, but in practice detonation forces impurities into the metal and makes repair very difficult – it may be worth considering replacing the head in a severe case. (Remember when buying another head that the early 43.5mm inlet valve head will fit the 84mm bore TCs and is worth more power – up to 10bhp in the 2*l*.) Also check for scoring in the cam box mating faces, and remember to use some silicon gasket on the rebuild if this is the case.

Check the condition of the valve heads, stems and tips. All the 8v TCs with the possible exception of the 124 1608 (BC) use nickel steel chrome-stem valves (not 21-4n) although the exhaust valves have 21-4n heads friction-welded to the steel stem. Because of the chrome plating, the stems do not tend to wear (the guides do), but the tips should be 'dressed-up' with a valve grinding machine if dished or peened and the buckets may also need renewal – look for good secondhand ones. Provided the valve heads are not cracked, impact damage can be removed (if they are not bent) by refacing in the valve grinder.

Check that the valve size is correct for the head – not as silly as it seems because early 42.4mm valve heads only had about 0.25mm wall thickness on the seat insert peripheral to the valve head and later valves were reduced to 41.8mm. Ideally there should be at least 0.5mm to prevent cracking. GCT have seen 'big valve conversions' where the valve is actually larger than the insert – which rather defeats the object of the conversion because the valve shrouds the seat at low lift.

Guides

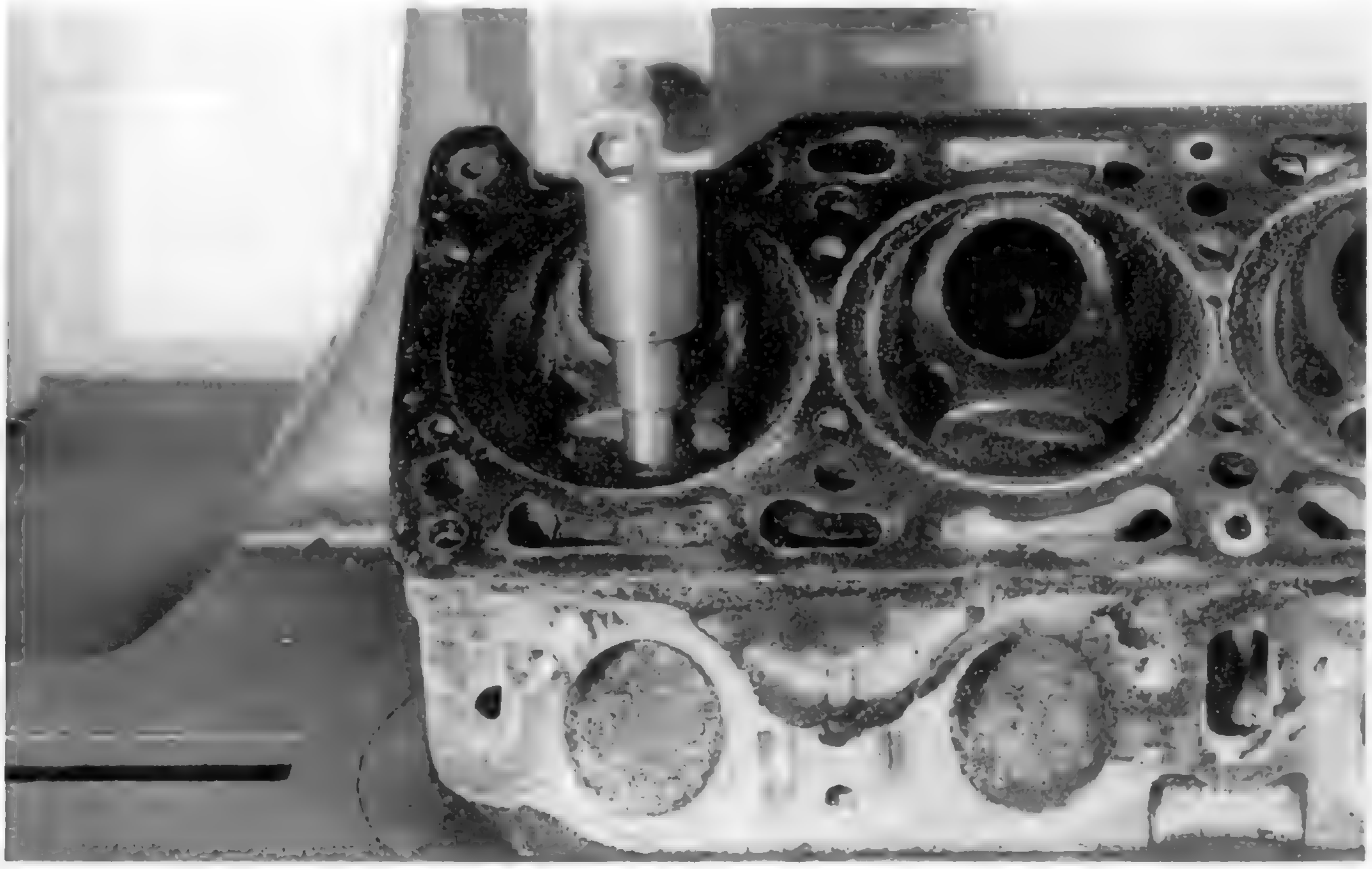
Cast iron guides are only available from Fiat in oversize, *ie* 14.05 on later 8v heads, so they need to be turned down to size to fit **(5/12)**.

Recommended guide dimensions should be:

Early	OD	15.02 ± 0.005mm (free)
	ID	8.000mm (free)
Late	OD	14.02 ± 0.005mm (free)
	ID	8.000mm (free)

These sizes will usually ensure a good fit with the head, but reaming back to 8mm may be needed as they may close up during fitting. Check the fitted guides

5/11: Wurth 'Time Sert System' for thread repairs. Tap (left) centres on old plug thread and opens out to larger diameter. Spotface tool (centre) relieves top of hole to ensure plug seat is at 90deg to hole axis. Spreader at right locks insert into place. Also available for 8mm and 6mm sizes.



5/12: Valve guides being pressed out of old head prior to soak overnight in decarbonizing fluid. Alternative to press is use of Fiat special tool and heavy hammer – perfectly satisfactory. Note that guides are pressed from inside-out on removal, from outside-in on fitting. Although head appears a hopeless case, it cleaned up perfectly. Watch out for factory oversize guides – keep old ones and measure OD. Oversize guides are not always available in odd sizes. Head is normally marked with oversize under valve spring seat washer (eg +0.05). Heads will accept up to 49mm inlet valve (biggest ever used at GCT) – forget offset guides. Always replace guides with bronze types (Fiat silicon/bronze or Colsibro) before porting – cast iron guides are prone to crack if shortened.

with a reamer. The OE aluminium-bronze guides supplied, for example, for the Lancia Delta Turbo 1600 (exhaust only) usually fit straight in the late TC heads, which saves a lengthy reaming process (bronze is very hard to ream – the best way is with a high-torque drill and cutting fluid). The interference between the guide OD and the bore in the head should be 0.0008"–0.0026", but the reader will readily appreciate that if the guides are not properly designed they will close up excessively on fitting. Finish-honing with Flex-Hone improves oil retention and increases their working life quite dramatically.

The running clearance between the valve stem and guide (8v) should be 0.0012"–0.0026"; certainly on the inlet side, the less the better. Early 16v turbo heads, however, suffered from insufficient stem-guide clearance, so check before purchase that the valves are not damaged, if buying a secondhand unit.

Valves

Fiat quote the following valve head angles (8v and 16v):

Seat 45deg ± 5min
Valve 45deg 30min ± 5min

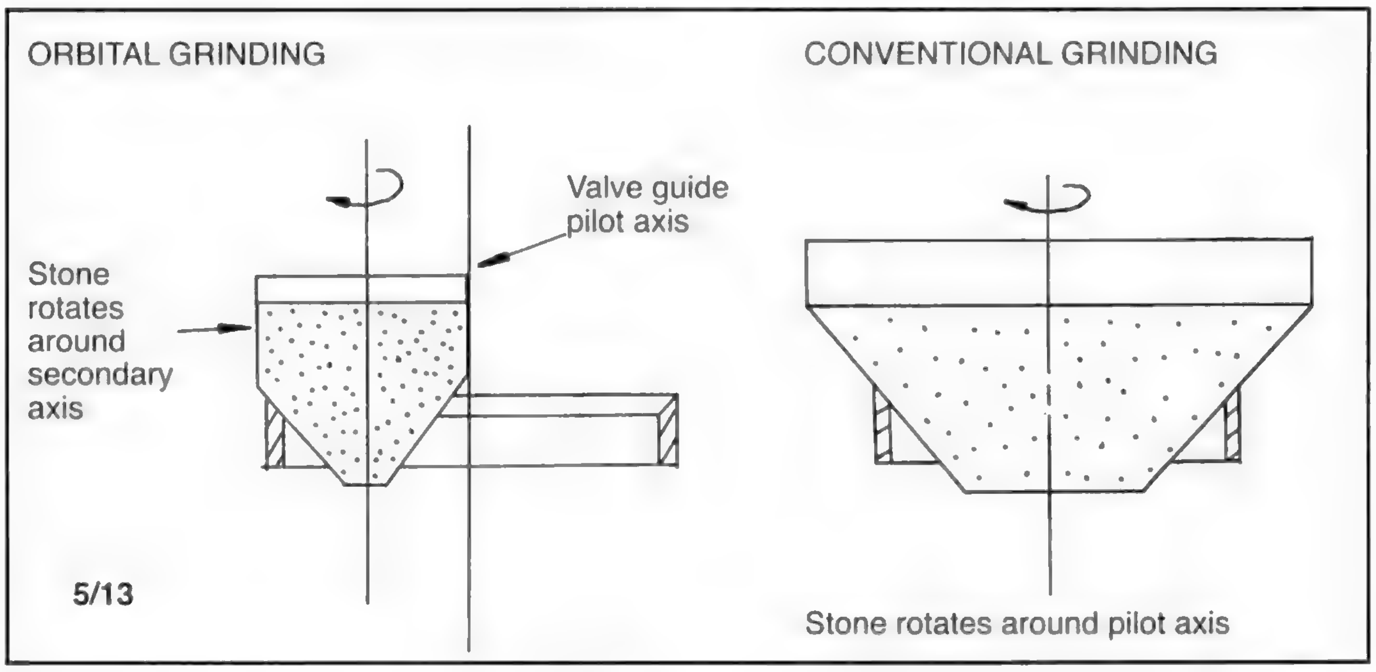
The ½deg difference creates an interference angle which gives better sealing when hot than 45deg-45deg. GCT have always preferred 46deg seat, 45deg valve.

The 8v stem diameter should be 7.974–7.992mm; if the stem is badly worn, replace, otherwise have them lightly refaced in a valve grinding machine. If there is no damage to the contact face (original 8v exhaust valves were stellite-coated to resist wear) the valves will lap in OK, but refacing is an easy way to check whether any are bent. A valve with a mild degree of distortion can be 'trued-up' on the grinder.

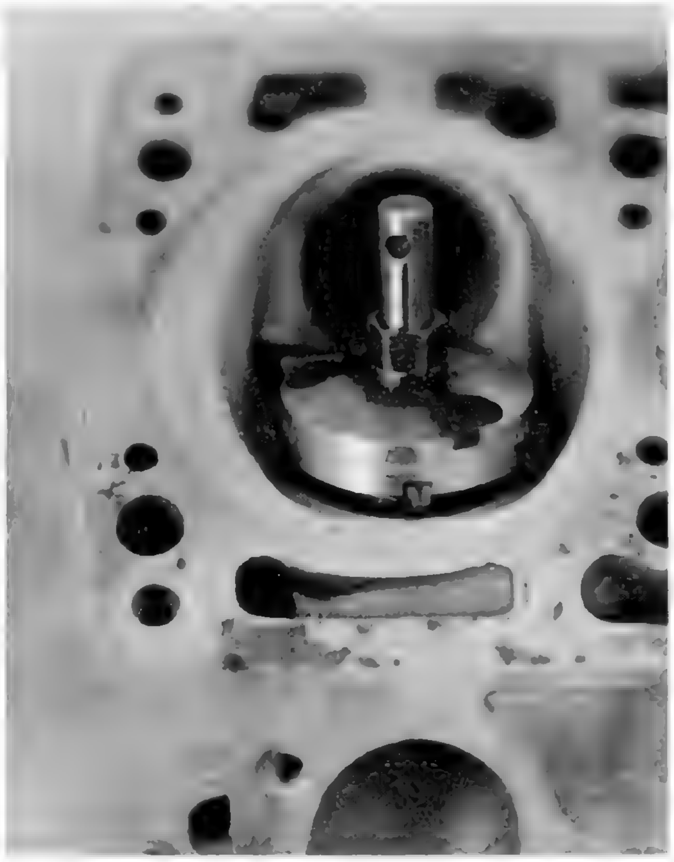
Seats

If guides are replaced the seats must be recut. There are two basic ways of doing this: grinding and cutting. Grinding is particularly useful when hardened seats are used (US and some late – *eg* turbo – models) but cutting is otherwise far quicker. The best grinders use an orbital motion whereby the stone rotates around the central pilot on its own axis (**5/13**).

Stones, of course, have to be accurately dressed. GCT use the US Neway cutters (sold in the UK by Sykes Pickavant) which replaces the conventional stone (see diagram) with a tool holder and a series of tungsten carbide cutters; 46°, 20°, 60° and 70° cutters are available to suit the various valve angles employed.



5/13



5/14: Neway 46/20deg cutter in place on pilot. Ratchet wrench can be used to turn, alternatively a power version is available. Check cutters and tool regularly for wear.

5/15: Some people do not approve of valve lapping: it has always been used at GCT. Chemico paste is one of best. Valve is standard Fiat inlet with approx 30° blending cut. Engineers' blue can be used for final check, plus of course vacuum or pressure test meter to check sealing. (With blue – put a small amount on valve contact face at three points and rotate on seat – should show up evenly on seat.)

Cutting is by hand (although a motor-driven option is available) and, as with the other methods, various pilot diameters are available. Neway cutters will cut all seat materials, although cutting fluid should be used on steel. For rapid initial removal, the Neway throat cutter (60 and 70° blades with a coarser profile) can be interchanged with the normal fine-profile seat cutters, which is a great help when cutting steel seats (**5/14**).

Neway cutters have the advantage that they keep their shape for literally dozens of heads and the tool form produces no radial deviances from the required figure (as can happen with stones). Pilots for all types should be maintained in good condition because they ensure accurate concentricity of the seat with the guide.

Single-point machine cutters are extensively used these days, especially with 16v heads. These normally employ a self-centring pilot and are notably quick and effective to use. As with other methods, the concentricity of the finished seat can be checked with a special dti, which centres in the guide bore if there is any doubt as to its accuracy.

A modest reconditioning job would only require light touching-up of the seats to the desired angle rather than the fully blueprinted type described later (**5/15**).

Head refacing

This may be carried out by grinding (on a surface grinding machine, using plenty of coolant) or by milling, using a multipoint cutter or single-point tool. GCT usually use a carbide-tipped tool, which will also cut through protruding inlet seat inserts without undue scoring. When using a mill, GCT tilt the cutting head fractionally to allow about 1–2thou" clearance under the tool trailing edge – this improves the surface finish. It is vital not to make this angle excessively large or the tool will 'scallop' the head face.

For refacing, the 8v head should be set up using the machine setting surfaces on the top side (between the cam boxes); this ensures that the head face will be accurately set above the valve inserts (one particularly bad example encountered at GCT required 18thou" to be machined off to bring it back to true) which, apart from anything else, ensures that all the combustion chambers will tend to be the same size. The 16v head may be set up on the top since it is accurately parallel to the head face.

To avoid problems with piston-valve clearances, only the minimum should be removed from the head. Detonation damage to the head face is very often worse than it looks, and a refacing operation of 3–12thou" will clean most TC heads. As a general rule, it is safer and more effective to raise the compression ratio by replacing pistons with higher domed models than refacing large amounts off the head. Even machining 20thou" off the 2*l* 131 head will only raise the CR to around 9.1:1–9.2:1, which is barely measurable on the dyno, and it is worth leaving an allowance on the head for future machining.

PORTING AND BLUEPRINTING

Part 1: THEORY AND FLOW TESTS

Porting is the generic name applied to modifications to the inlet and exhaust ports of the engine; blueprinting refers to the work required to enable the cylinder head design, including the valve area, to conform precisely to the designer's original layout.

The design of the TC 8v cylinder head is exceptionally effective and is the main reason behind the success of the engine as a competition unit, but any serious attempt at raising the output should give consideration to this work since there are finite limits to what can be achieved from the standard item.

The normally aspirated engine relies principally on a small pressure differential between the vacuum in the cylinder and the inlet tract to work at all, since only atmospheric pressure is available to force mixture into the combustion chamber. The greater this pressure difference the more power will result, but that is not the whole story. There are a series of complex, interactive factors involved, which will now be dealt with.

Flowrate and discharge coefficient

The theoretically perfect (*ie* maximum possible) flowrate through a valve throat is identified by the equation:

$$Q_{th} = 21.8d^2\sqrt{H}$$

where

Q_{th} = flowrate cu ft/min
d = valve throat diameter (in)
H = pressure drop across throat (in H_2O)

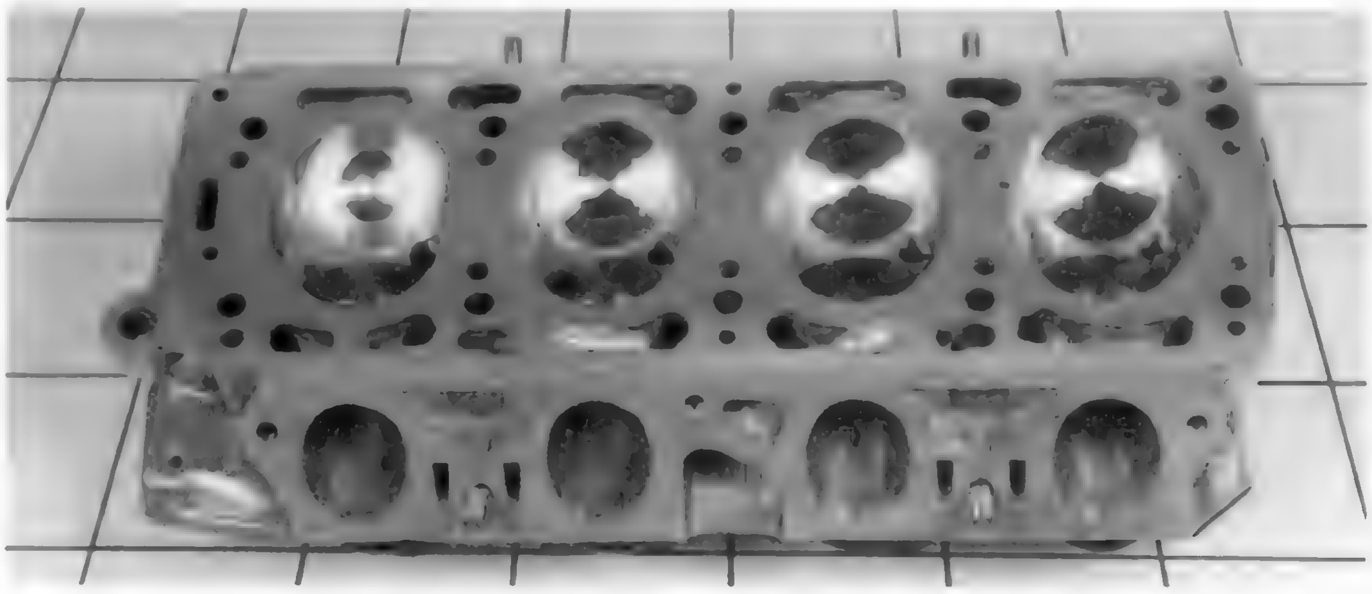
Hence, for a pressure drop of 10" H_2O (a figure commonly used in airflow testing) with a standard 41.8mm valve TC head (37mm throat) the theoretical flowrate is given by:

$$Q_{th} = 21.8\left(\frac{37}{25.4}\right)^2\sqrt{10} = 146\text{cu ft/min}$$

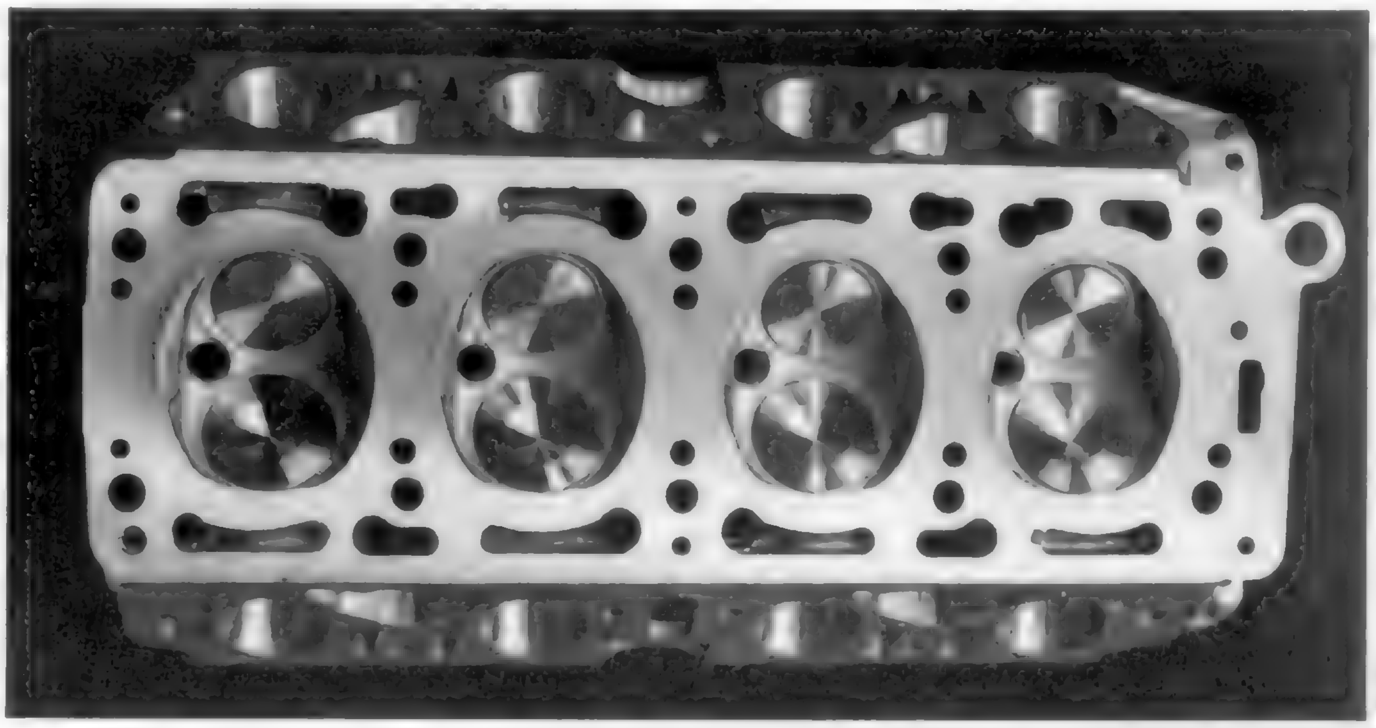
To compare flowrate at 10" H_2O with, say, a head tested at 25", multiply 10" flowrate by 1.58 (the sq root of 25 divided by 10).

The actual flowrate will be less due to losses caused mainly by the intrusion of the valve into the airstream (**5/16, 5/17**). This is expressed by application of a discharge coefficient C_d, where the value of C_d depends upon the lift/diameter ratio ($^l/_d$) of the valve/valve throat. In other words, a valve lift of 10.4mm with a valve throat diameter of 37mm will give an $^l/_d$ value of $^{10.4}/_{37}$, *ie* 0.28. In order to derive values of the discharge coefficient C_d for the TC, it is necessary to first carry out airflow tests on the head. From the figures of H (pressure drop, in H_2O) and valve throat dia (in) it is possible to calculate C_d.

For example: Using the figures from Test 1 (Graph 1), the $^l/_d$ ratio and flowrate through the inlet tract with a 40mm valve throat (44mm inlet valve, blueprinted seat) at various lifts are as follows:

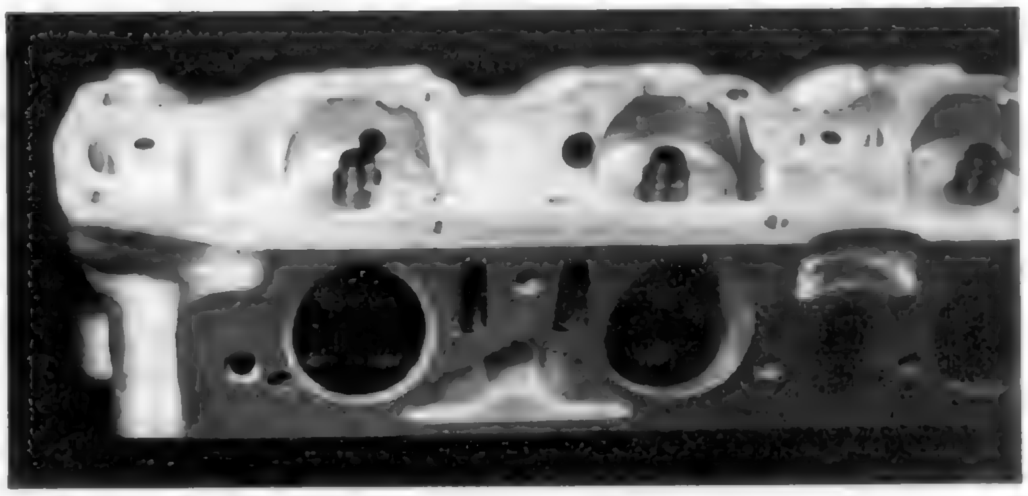


5/16: 49/40 big-valve head. Large ports gave high flowrate, but due to shrouding and low port velocity initial torque results on n/a engine were disappointing. A head of this type is best suited to a pressure-charged application. Coolant galleries were welded up on this model, allowing removal of 3mm from base of port.



5/17: Blueprinted, ported 130 TC head with 44/38 race valves. Note unmodified combustion chamber. Only necessary to deshroud chamber when 45 or larger inlet valves are used. Combustion chamber design is late pattern used on 124 Sport 1800, Beta 2l Delta/Prisma 1600, 105/130 TC. This head is good for around 188bhp, 150lbf ft torque, depending on cams and CR (45s-40 choke). Proximity of inlet valves to head face is not unusual – bigger valves (depending on how much is milled off head) can protrude beyond head face. (Check radial clearance in this case.) Similar GC design with 44/36 valves was used on flowbench Test No 1.

Lift (mm)	$^L/_d$	Actual flowrate Q (cu ft/min)	% increase in flow
4	0.11	49	
5	0.14	60	22%
6	0.16	70	17%
7	0.19	81	16%
8	0.22	91	12%
9	0.24	100	10%
10	0.27	105	5%
11	0.3	109	4%
12	0.32	113	3.7%
13	0.35	117	3.5%



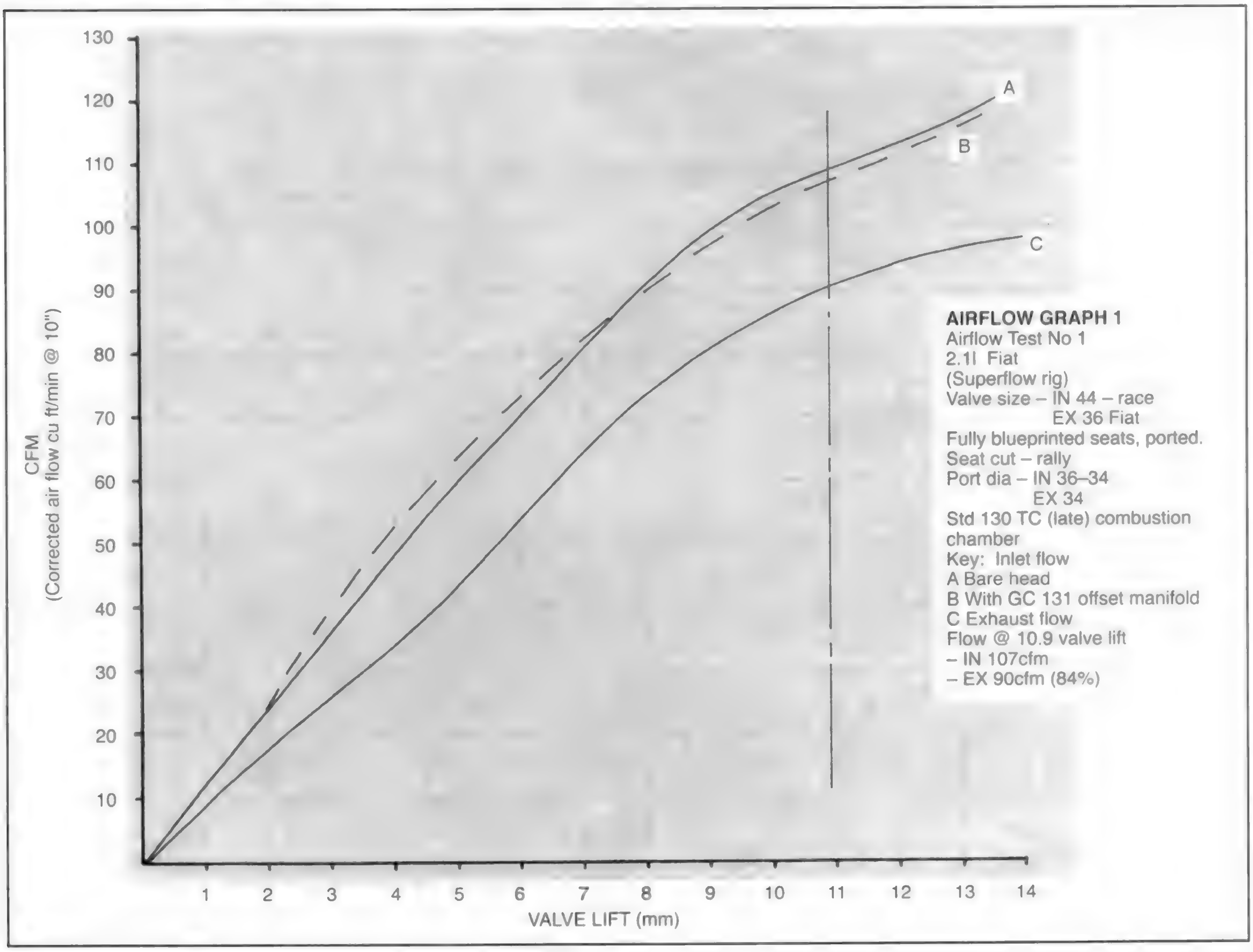
5/18: 49/40 2l Lancia Beta head showing enlargement to inlet ports. Exhaust port dia is 36mm throughout. Inlet port is 40ø at port face tapering down to 38 at port venturi. Wall thickness is approx 3mm on both ports (see flowbench Test No 3).

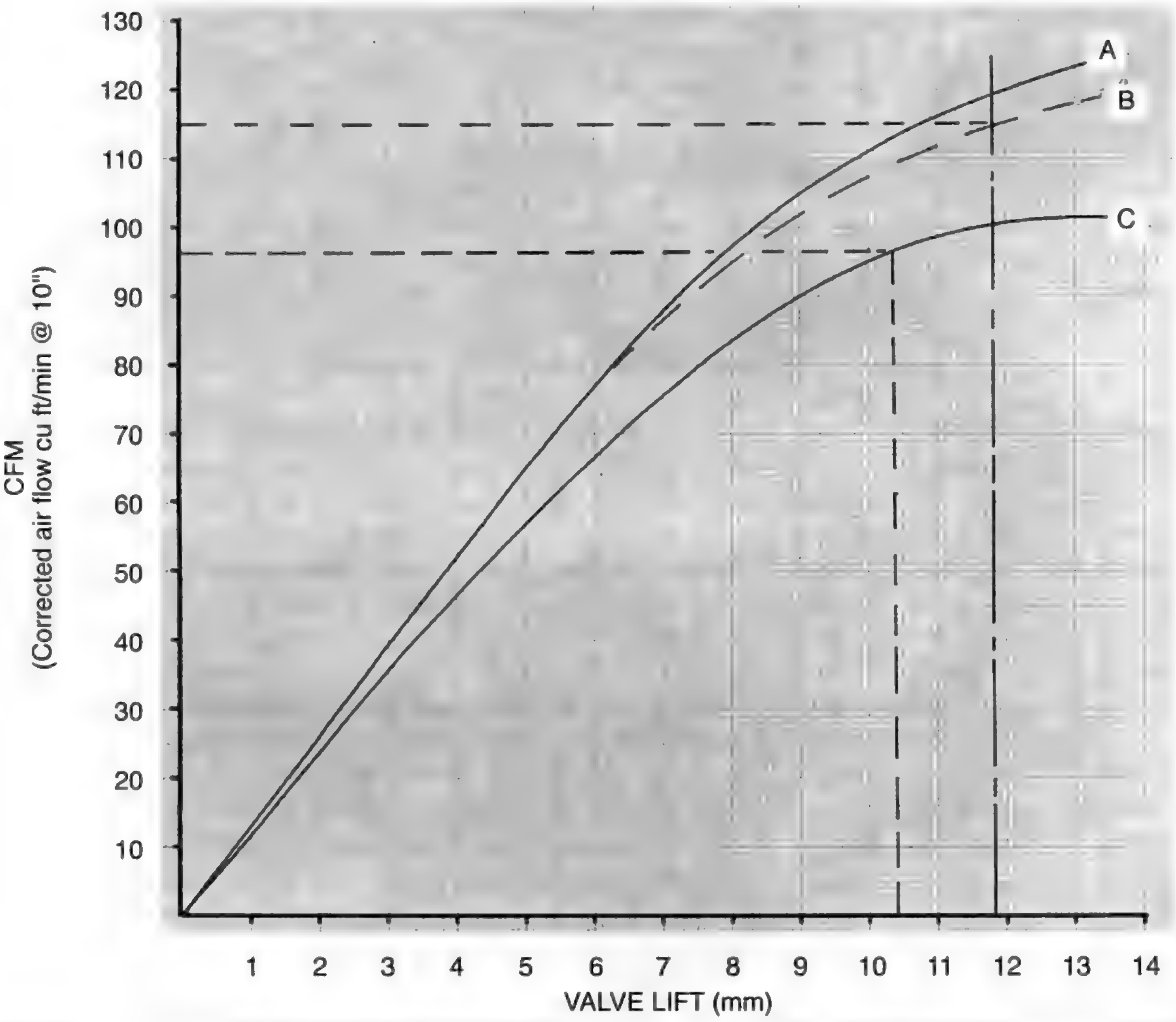
Note: Airflow figures are 'corrected' to atmospheric conditions.
The relationship between theoretical flowrate and actual flowrate is given by:

$$Q = Q_{th} \times C_d$$

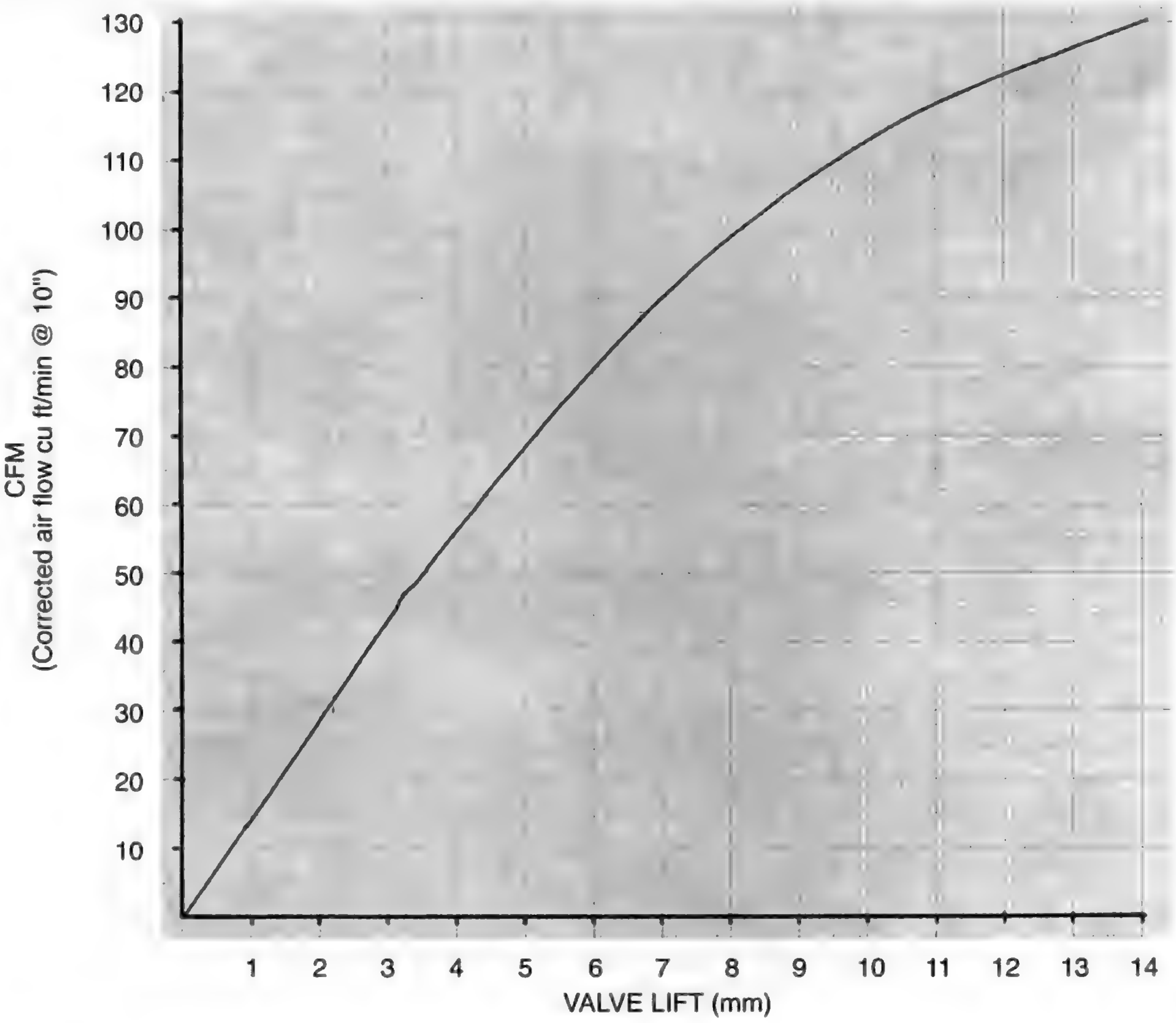
Therefore, as Q_{th} and Q are known, C_d can be calculated, *eg* at lift 8mm above, flowrate = 91 cfm, hence $91 = 146 \times C_d$; $C_d = 0.62$.
Graph 5 shows C_d plotted against $^L/_d$ ratio for two heads with 49mm and 44mm inlet valves: it is worth noting at this stage that above $^L/_d = 0.26$, the percentage increase in flowrate starts to diminish.

Good race head design (5/18) is a question of achieving a balance between the law of diminishing returns from this effect, and the fact that from the above equation, if the valve throat size is increased the flowrate should increase! Graphs 1–3 show the true flowrate achieved with a variety of valve, valve throat and lift conditions, and it is clear that the big valve wins throughout the range for a given lift, particularly over 10mm.

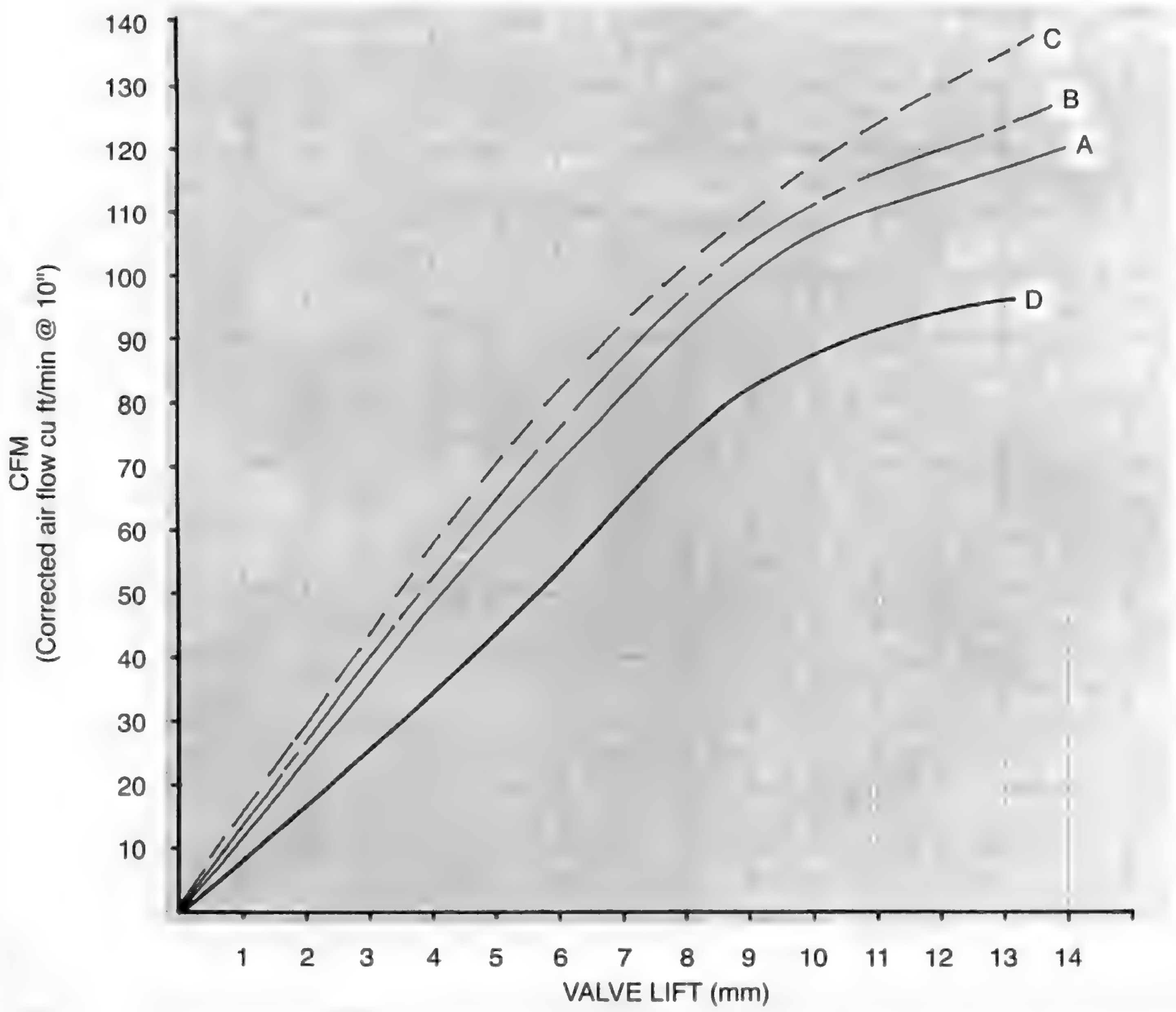




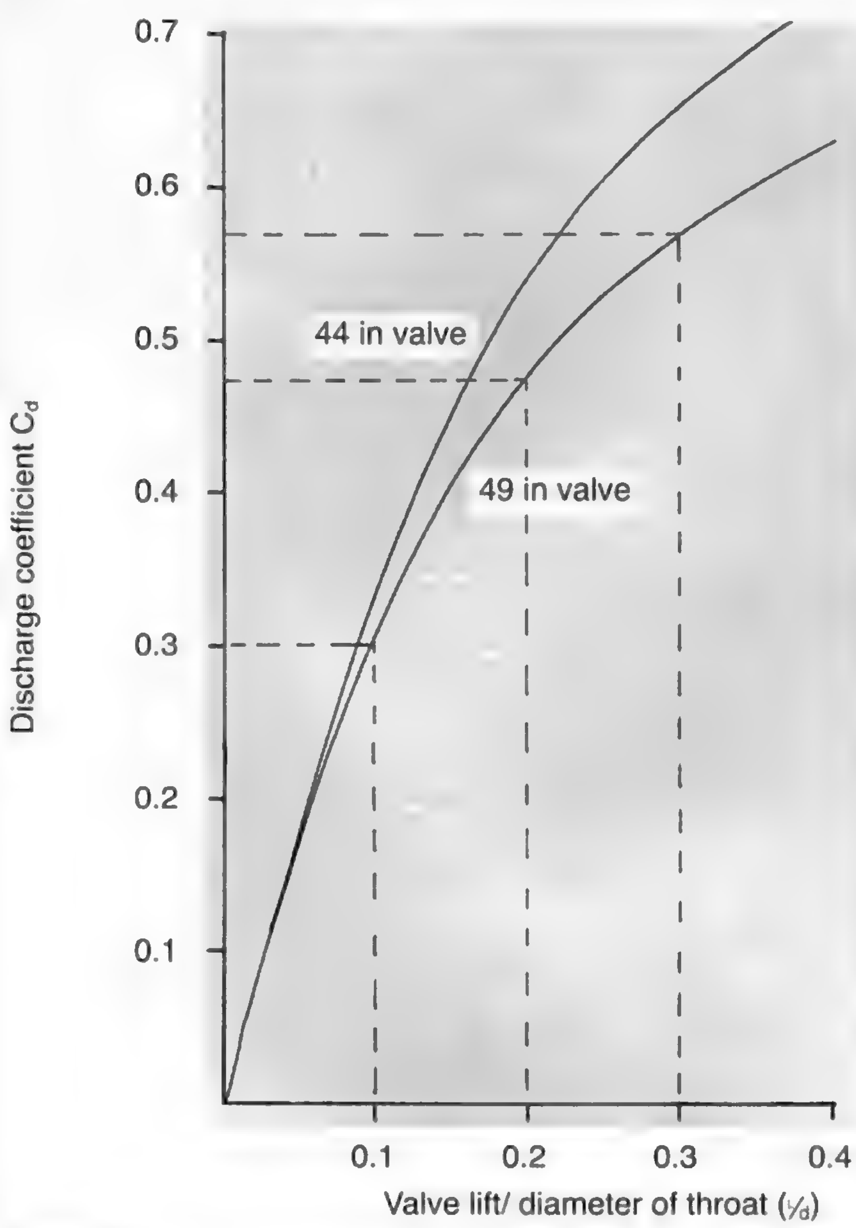
AIRFLOW GRAPH 2
Airflow Test No 2
(Superflow rig)
2.1l Fiat
Valve size – 46 IN – race
40 EX – race
Fully ported/blueprinted
Seat cut – Endurance race
Port dia – IN 38–36
EX 36
131 2l (late) modified combustion chamber
Key: Inlet flow
A Bare head
B With GC S/S manifold (1¾" long) 48 DCOE (40 choke)
C Exhaust flow



AIRFLOW GRAPH 3
Airflow Test No 3
(Superflow rig)
2.1l Fiat
49mm inlet valve
Seat cut: ultimate race
Port dia 40–38
48 DCOE (40ch) fitted
Late 131 modified combustion chamber (deshrouded only)



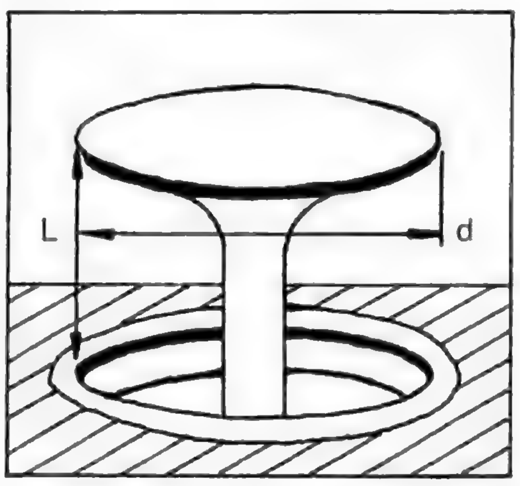
AIRFLOW GRAPH 4
'Barehead' airflow results
Superimposed (from Tests 1–3)
A 44 IN valve
B 46 IN
C 49 IN
D 36 EX



GRAPH 5
C_d versus $^L/_d$ ratio for 2 blueprinted/ported TC heads

Curtain area

The reason for the diminishing effectiveness of raising the $^L/_d$ ratio beyond, say, 12mm on the TC is that at some point where the annular area under the valve (curtain area) is greater than the throat area, increasing valve lift produces decreasingly significant increases in airflow (**5/19, 5/20, 5/21**); *eg* 40mm throat (44mm valve) – see table below:

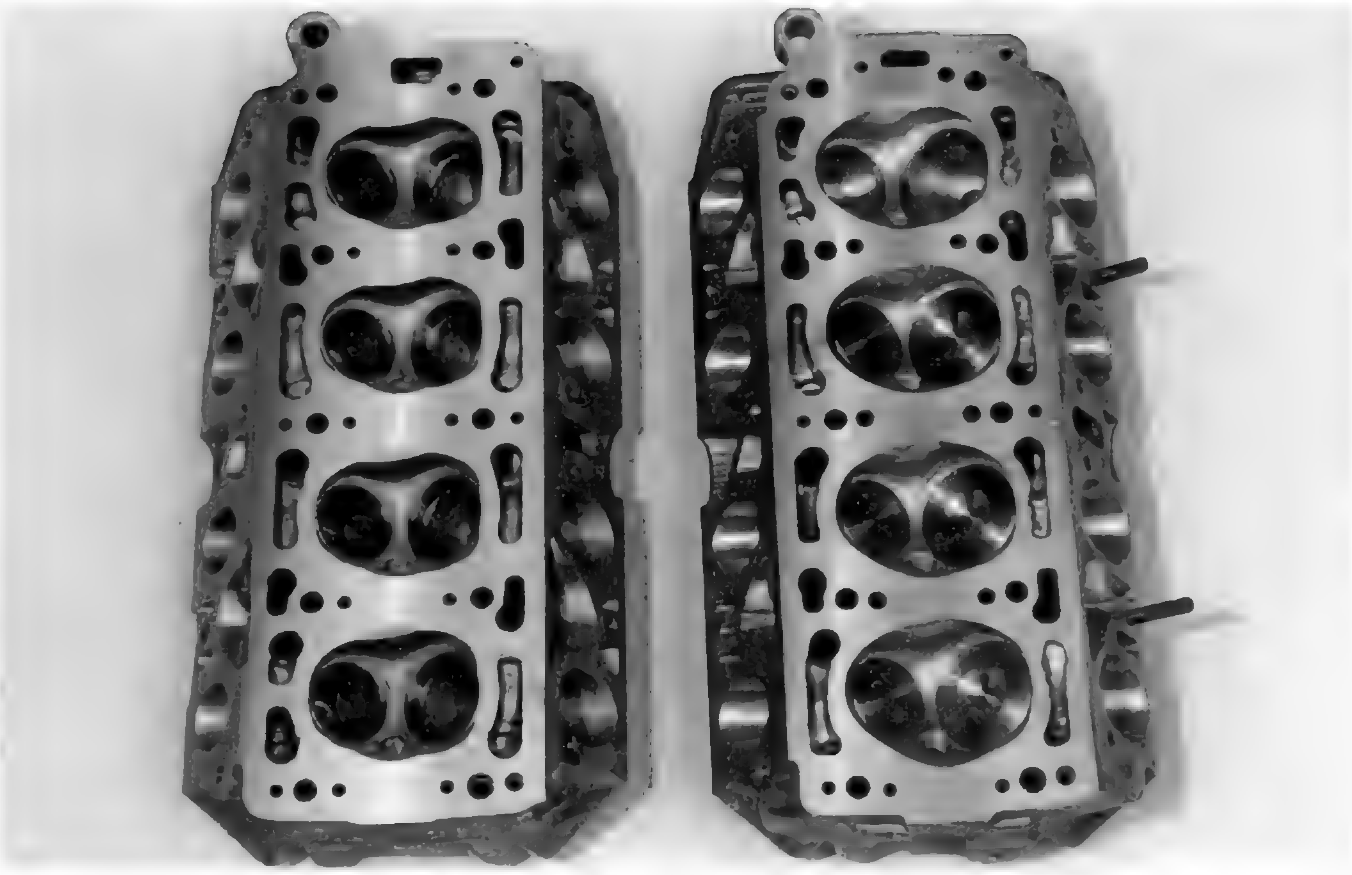


5/19:
Annular (curtain) area
$= \pi \times d \times L$
Valve throat area (appr)
$= \frac{\pi d^2}{4}$

L (mm)	$^L/_d$	Annular area (mm^2)	Valve throat area (approx) (mm^2)	% by which annular area is greater
10	0.27	1256	1256	=
11	0.3	1375		9.5%
12	0.32	1500		19%
13	0.35	1625		29%

It may reasonably be assumed from the flow test results that gains on a race engine may be expected well beyond the industry standard of $^L/_d = 0.26$, *eg* up to $^L/_d = 0.35$, *ie* where the curtain area is 29% bigger than the valve throat.

It is possible for a small inlet valve size to achieve high flowrates at high lift and thus raise torque. But in order to increase the low-lift flow (and hence broaden the spread of torque) it is safe to assume that only a bigger valve will do this. This is not to say that big valves are entirely without penalty: firstly there is the question of the engineering expertise required to fit them, the cost of the valves themselves plus, significantly, the fact that not only does a big valve allow more 'flow' in, it also allows more 'reverse flow' around TDC, so clearly the lift at TDC of the camshafts must be taken into account.



5/20: Very early 125 head, left (Fiat Samantha). Seats have been blueprinted, ports unmodified. Head at right is 49/40 race version with modified combustion chambers. Similarity in design is no coincidence – same basic pattern was retained when blocks were stretched from 80–84mm bore.

Pressure drop and velocity

In theory, using the equations:

$$Q_{th} = 21.8d^2\sqrt{H}$$

$$\text{and } Q = Q_{th} \times C_d$$

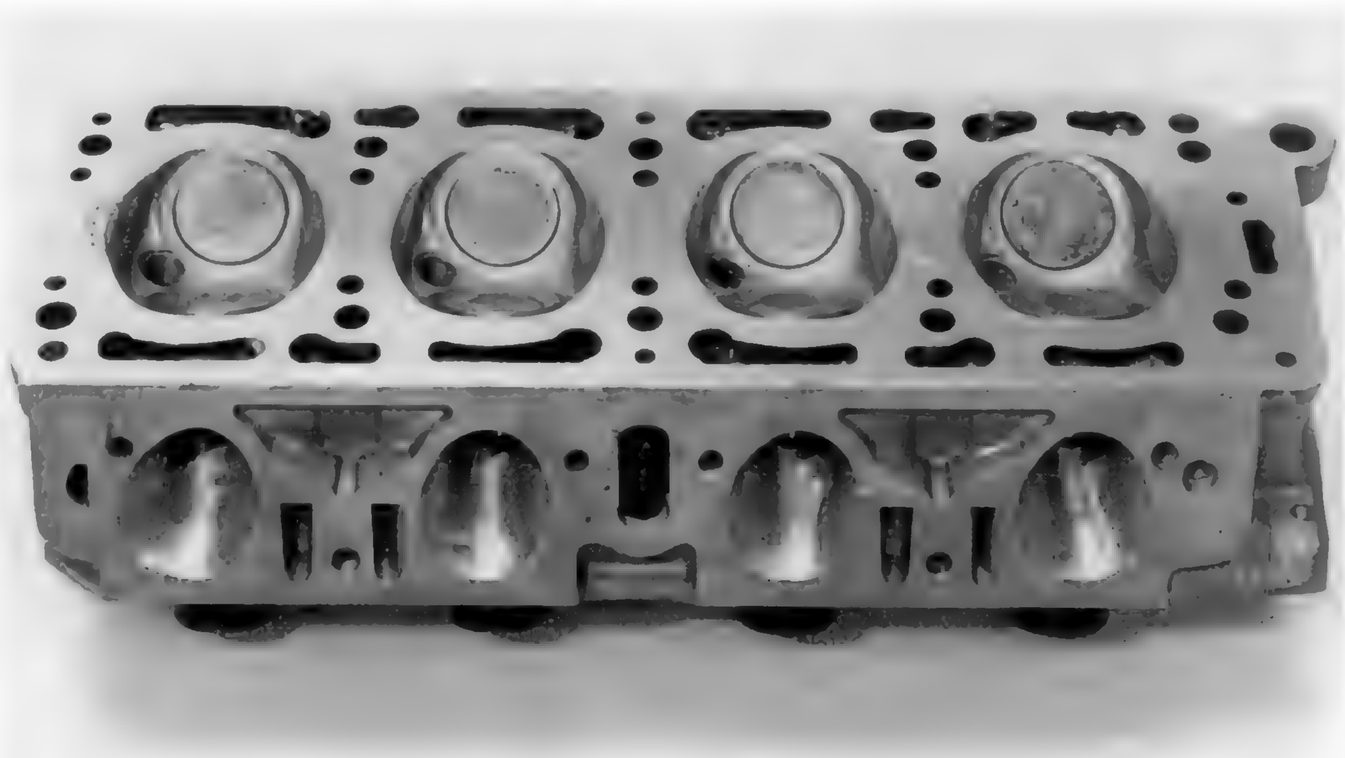
it should be possible to make flowrate predictions by extracting values of C_d from Graph 5. In practice, this is not so simple, because as seen from the results of $^L/_d$ *vs* C_d, the actual value of C_d obtained under empirical conditions varies with the valve size and the differing valve/seat designs used. There is also the question of port velocity. The velocity through the port varies as the square root of the pressure drop H – precisely one of the key components in the above equation.

Measurement of air velocity through a port using an orifice plate meter is given by the equation:

$$V = 66.7\sqrt{H} \text{ (ft/sec)}$$

this leads to

$$H = \left(\frac{V}{66.7}\right)^2$$



5/21: Late 131 2l head blueprinted/ported with reconditioned standard valves. Curious shape of inlet ports is due to matching job with early (non-GC) inlet manifold. Amount of metal removed indicates chronic mismatch that existed.

Thus a high-velocity port will give a high pressure drop and *vice versa*. If the port velocity increases, so does the pressure drop (H), and using the above equation, the flowrate (Q) should also increase. The velocity in the inlet port is governed principally by the piston speed, *ie*:

mean piston velocity $$V_p \text{ (ft/sec)} = \frac{\text{stroke (in)} \times \text{speed (rev/min)}}{360}$$

or in other words: $$V_p = \frac{Sn}{360}$$

mean gas velocity $$V \text{ (ft/sec)} = \frac{Sn}{360} \times \text{ratio } \frac{\text{bore area}}{\text{valve throat area}}$$

hence: $$= \frac{Sn}{360} \times \frac{D^2}{d^2}$$

(where D is bore dia, d is throat dia)

ie: $$V = \frac{Sn}{360} \times \left(\frac{D}{d}\right)^2$$

So, immediately a number of new parameters become relevant:
Bore size (D)
Stroke (S)
Engine speed (n)
Hence it is obvious that for the same values of n, D and d, the 2*l* engine with its longer stroke will give a higher mean port speed, a higher pressure drop across the throat, and more flow than, for example, the 1585.

Port size, although ignored in the equations, is also important. A small throat will flow the same mass/sec of air as its adjoining port (Bernoulli's theorem), though if the port is larger than the throat, the port speed will be lower; if the port size is too large, the air speed will be too low and momentum will be lost. For optimum performance, the air momentum must be maintained.

There are a number of ancillary factors, which will also be discussed later, which affect airflow, and it may be seen readily that the science of airflow prediction is highly complex – one of the reasons for the increasing use of actual flowbench testing in the industry!

BHP prediction

Superflow flowbench data predicted the following bhp results from the flow tests on Graphs 1–3:

Valve size (mm)	Predicted output (bhp)	Actual output (bhp)	Engine spec (2*l*)
44	177	186	IIIA cams 45 DCOE (38ch)
46	189	200	IIID cams 45 DCOE (40ch)
49	209	Not confirmed at time of writing	–

In other words, the actual outputs were between 5%–10% higher than the flowbench tests, and indeed the engine specs cannot be regarded by any standards as the ultimate for the given valve sizes. One might conclude therefore that the primary advantage of the flowbench is to indicate whether mods to the head might lead *directly* to a flow increase, whereby a power increase might reasonably be expected; actual prediction of absolute bhp is less certain. A pressure drop of 10" H_2O represents:

$$V = 66.7\sqrt{H} = 210 \text{ ft/sec}$$

The results from dyno tests would tend to indicate that the port velocity of the TC can be significantly higher. It would be nice if it were possible to substitute higher values of H into this equation and estimate the flowrate from various heads at different values of C_d, L and d. However, probably due to the multiplicity of other factors involved, this does not work particularly well in practice. In conclusion, 10" H_2O is a satisfactory figure for undertaking back-to-back flow tests; higher H values may well give more impressive flow figures on paper, but definitive power outputs can *only* be obtained from dyno testing.

Thoughts on valve size – the influence of cam lift

One particularly useful conclusion that may be drawn from flow testing is the exhaust valve characteristic required. As a general rule of thumb on normally aspirated engines, the exhaust valve (and port) should be able to flow at 75% of the peak inlet flow. This was used to good effect in *Case History No 2*. The early engine used 46/40 valves with 11.8 (actual) lift cams.

From Graph 2, the peak flowrate through the exhaust with this set-up was 87% of the inlet, *ie* too high, and indeed the torque characteristic below 5000rpm was very poor. Swapping to a 10.4 (actual) lift cam immediately enhanced the torque below 5000rpm very substantially. Of course, this is not entirely attributable to a reduction in the peak exhaust flowrate (at exhaust valve opening, less gas pressure would be lost from the lower-lift cam), but also because with the milder cam the lift at TDC (and hence reverse-flow tendency) was reduced. Nonetheless, this particular flowbench result was instrumental in redesigning this GCT 2*l* Fiat, and initial race results have showed that enhancing the torque between 4–5000rpm was a viable move.

On turbo/supercharged engines, because the fresh charge is being forced in at above atmospheric pressure and the mass of charge is greater, one might expect to use a larger exhaust valve (or more lift) than with a normally aspirated engine, but results for these engines obtained at GCT suggest that, for various reasons, this is not necessary.

It was similarly noticed *(in Case History No 2)* that, although the high peak lift inlet cam gave good torque at 6000–7000rpm, reducing inlet cam lift at TDC (Tests 3, 4) gave a much better mid-range torque response at the expense of only about 5% of the original 6000rpm figure. It was concluded that the high lift at TDC-big valve combination was causing serious low-speed reverse flow of exhaust gas into the inlet port. The final LATDC was 4.5 inlet compared with 5.5mm.

Peak lift should be as great as mechanically possible for optimum torque. Additionally, because of the large opening area of the big valve, later opening sustains the vacuum condition in the cylinder, leading to a higher port velocity.

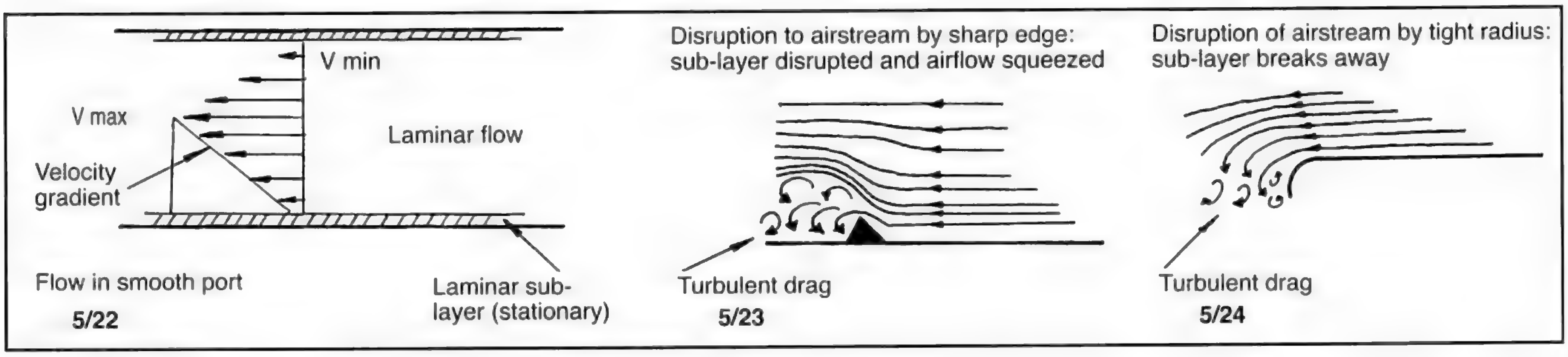
Port finish and shape

All the flowbench figures were obtained from heads with a 'wet' 80 grit finish (using an 80 grit Ataband lubricated with aerosol oil). There is nothing to suggest from the dyno results that improving the port finish would yield substantially more torque, indeed the reverse may result. The velocity in the inlet is not the same across the full diameter of the port. Velocity is a maximum in the centre of the port in the parallel (or near-parallel) section of the port, and reduces as the airstream (laden with fuel) approaches the port walls.

If the port finish is relatively rough, a static laminar sub-layer will cling to the wall and provide a slippery medium over which the airstream will flow. If the laminar sub-layer is lost because the surface finish is too smooth, the airstream will become turbulent and the flowrate will drop. It is worth noting that on *all* airflow and dyno tests recorded in this book the 'as cast' manifold finish was retained.

The inlet tract conforms to normal aerodynamic theory in the sense that wherever a sharp edge or tight turn is presented to the airstream, the flow may become turbulent, in which case drag results and flow is lost (**5/22–5/24**).

The effect of port finish in the exhaust port is significant only from the point of view that excessive intrusions will create localized pressure increases; polishing is completely unnecessary since the port will 'carbon-up' within minutes of running the engine. The exhaust gas is extremely hot and turbulent and will tend to create its own surface finish by the scrubbing action of the gas (the same phenomenon occurs with gas-operated firearm reload systems).



5/22 5/23 5/24

Combustion chamber modifications
Modifications may safely be carried out to the early-pattern and late reverse-port heads to improve flow by unmasking the valves.

However, on late-port heads (*eg* Beta 2*l* and 105/130 TC types) do not modify the combustion chambers unless fitting valves over 44mm, where the larger valve may be shrouded by the chamber walls. The valves should have about 4mm of clear area around their periphery.

Increasing the combustion chamber size may prove to be a trade-off against compression ratio, so ensure that the piston intruder dome is large enough to give the CR needed, or the net power may turn out to be less! It is also worth blending piston valve recesses to improve flow around TDC if CR considerations allow.

Carburettor and manifold losses
Interestingly, the 2¾" offset sidedraught GC bare manifold used on Test 1 yielded measurably more flow from 2 to 7½mm lift than the head, although it lost out above 8mm!. This may be due to greater momentum induced in the airflow by the longer lengths.

This was used to good effect in *Case History No 2* (although wave action may certainly be contributory). On Test 2, when a straight-shot manifold was used on the bare head (without carb) the result was +1cfm between 2.2–6.7mm lift, and –1cfm between 8.7–12.2 lift, with no loss over 12.2!

Carburettor losses were (%cfm):

45 DCOE 40 ch	6.8% (46 valve)
	8.5% (49 valve)
48 DCOE 40 ch	5.1% (46 valve)
	6.6% (49 valve)

(all figures with std rampipes)
Later tests showed that by careful blending of the internal shape of the carbs and knife-edging the throttle assembly, these losses could be reduced by around 50%.

Inlet tract momentum and carburettor airstream velocity
A long inlet tract creates a moving column of air with a certain momentum. If the momentum is high enough it can raise the flowrate through the throat when the inlet valve opens; the penalty is that if it is too long, the petrol droplets fall out of the airstream and 'pool' in the tract.

TC heads have tended to use venturi ports (see table of port sizes) where the airstream is fed through a venturi in the area of the valve guide. This tends to lower the air pressure and raise its velocity in the same way as a carburettor choke and helps to compensate for the flow loss created by the intrusion of the valve stem and guide into the port.

The same principles can be used in designing the inlet tract with a progressive taper (with an included angle of 14deg or less), indeed all GCT engines have been built with this in mind. This effect is especially useful to build up 'overpressure' behind the inlet valve (which is actually closed for over half the cylinder working cycle), in order to increase throat velocity when it opens.

Carburettor size (normally aspirated engines)
The size of carburettor and choke is critical. If the carb is too large, the airspeed will drop and may stall. If the step between the carb and choke is too great, the local airspeed will drop (the idea being to maintain or increase the airspeed throughout the inlet tract) and power will be lost. On a test, a GC engine with St III cams, 10:1 CR and 42/36 valves *lost* 30lbf ft torque using 48s (40 choke) compared with 45s (40 choke).

It is important to ensure, however, that the chokes are large enough to provide a sufficient airstream or the engine will run over-rich (for example, a 2*l* St II engine fitted with 45 carbs must use at least 38mm diameter chokes). Actual prediction of the carb size for a given engine is largely a question of experience. Based on extensive experiments at GCT, the following recommendations may prove useful:

1: **Choke size**
To minimize the velocity drop between the throat and choke, the valve throat diameter should be at least 95% of the choke size and may be up to 120% larger.

2: **Carburettor size**
For optimum velocity increase through the choke, the carb diameter should be 1.12–1.2 times choke size.

3: **Port venturi size**
The port venturi diameter should be 73–85% of carb size (ensure not more than 7° to horizontal taper). The gas velocity through the carb and choke may be estimated by using the equations:

TABLE OF SUGGESTED PORT DIMENSIONS (FROM GCT DYNO-TEST RESULTS)

Engine size (cc)	Cams	Inlet valve size (mm)	Inlet port venturi dia (mm) 'A'	Inlet port face dia (mm) 'B'	Manifold length (mm)	Carb choke size (mm)	Power potential (bhp)
2*l*	Std	42	32–34	35	2¾	45/36	165
	St II/III	44	34	36	2¾	45/40	176–190
	St III/IV	46	35	37	1¾–2¾	45/40 or 48/40	192–210
	St IV	49	35 (Parallel port)	37	1¾–2¾	48/40	NYK
1600 (1585)	St II	42	32	34	2¾	40/34	145
	St III	44	33	35	2¾	45/36	175

NOTE: BHP predictions depend on exact engine specification, *eg* cams, CR and exhaust design.

$$V \text{ carb (ft/sec)} = \frac{Sn}{360}\left(\frac{\text{dia bore}}{\text{dia carb}}\right)^2$$

$$V \text{ choke (ft/sec)} = \frac{Sn}{360}\left(\frac{\text{dia bore}}{\text{dia choke}}\right)^2$$

Part 2: PRACTICAL MEASURES (in order of precedence)

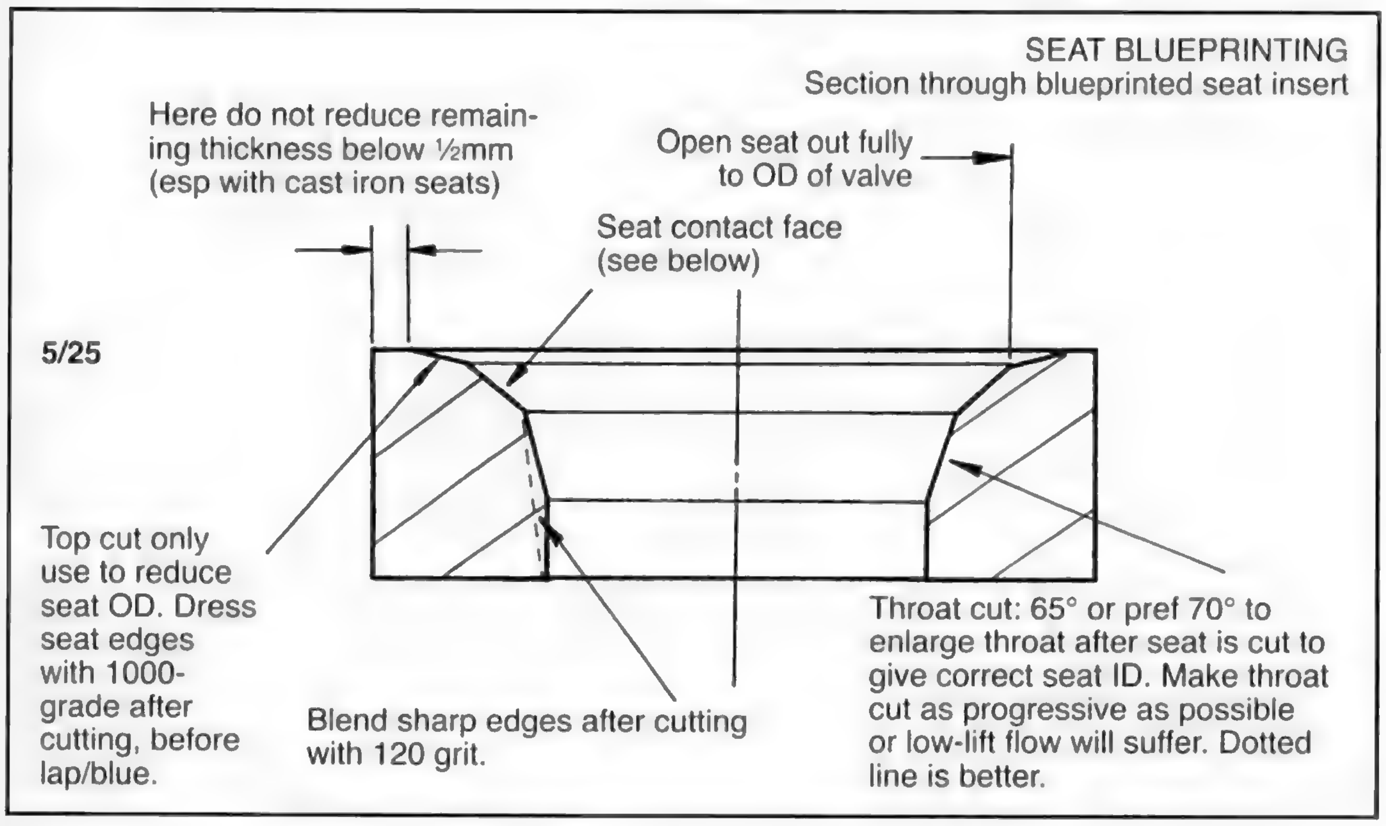
Seat cutting

Blueprinting the seats for more flow (and hence better torque/bhp) is a more advanced approach than the basic reconditioning cut described earlier. Good blueprinting improves the discharge coefficient of the valve throat area (**5/25**). Start by measuring the valve diameter. The OD of the contact face needs to be the same as this measurement. The seat insert must be larger than the valve, as shown, to avoid cracking or distortion.

If a lot of material has to be cut out to enlarge the OD, start by opening out the throat. (Surface finish and concentricity may suffer if cutting is attempted on a contact face wider than about 2mm.) Try to avoid a top cut by cutting accurately out to the valve head diameter first time. Once the OD is established, the throat can be opened out as shown, and finally, the various sharp edges blended with 220-grade emery (**5/26**).

5/26: Useful port/seat finishing tools. Top left to bottom right: Ata Scroll, Garryson Spiraband, Atabands 1cm, 2.5cm – 6mm bolt shank slotted with emery paper fitted. Neway cutters: left to right (top) 20/46deg (large and small), 30/60deg; left to right (bottom) 70deg, 60deg.

It is good practice to retain a mild taper in the throat (although flow is higher with a parallel one) since it enables the seats to be recut at the end of the season with metal 'in-hand'. To keep the flow laminar as long as possible, the throat should be as progressive as reshaping will allow – with very large valves (*eg* 49mm) this is almost impossible, which is the reason why the discharge coefficient is lower than for the 44mm valve.



ENGINE TYPE	SEAT DIMENSIONS INLET	EXHAUST	SEAT ANGLES
	THOU" PER INCH OF VALVE DIA		
Road / Rally	35	42	45–46° seat
Sodium-cooled ex valves	n/a	30	45° valve
Endurance Race	30	As Rally	
Ultimate Race	20	As Rally	

NOTE: 46° seat/45° valve gives 'interference' contact resulting in tighter seal when hot.

Valve modifications

A – inlets

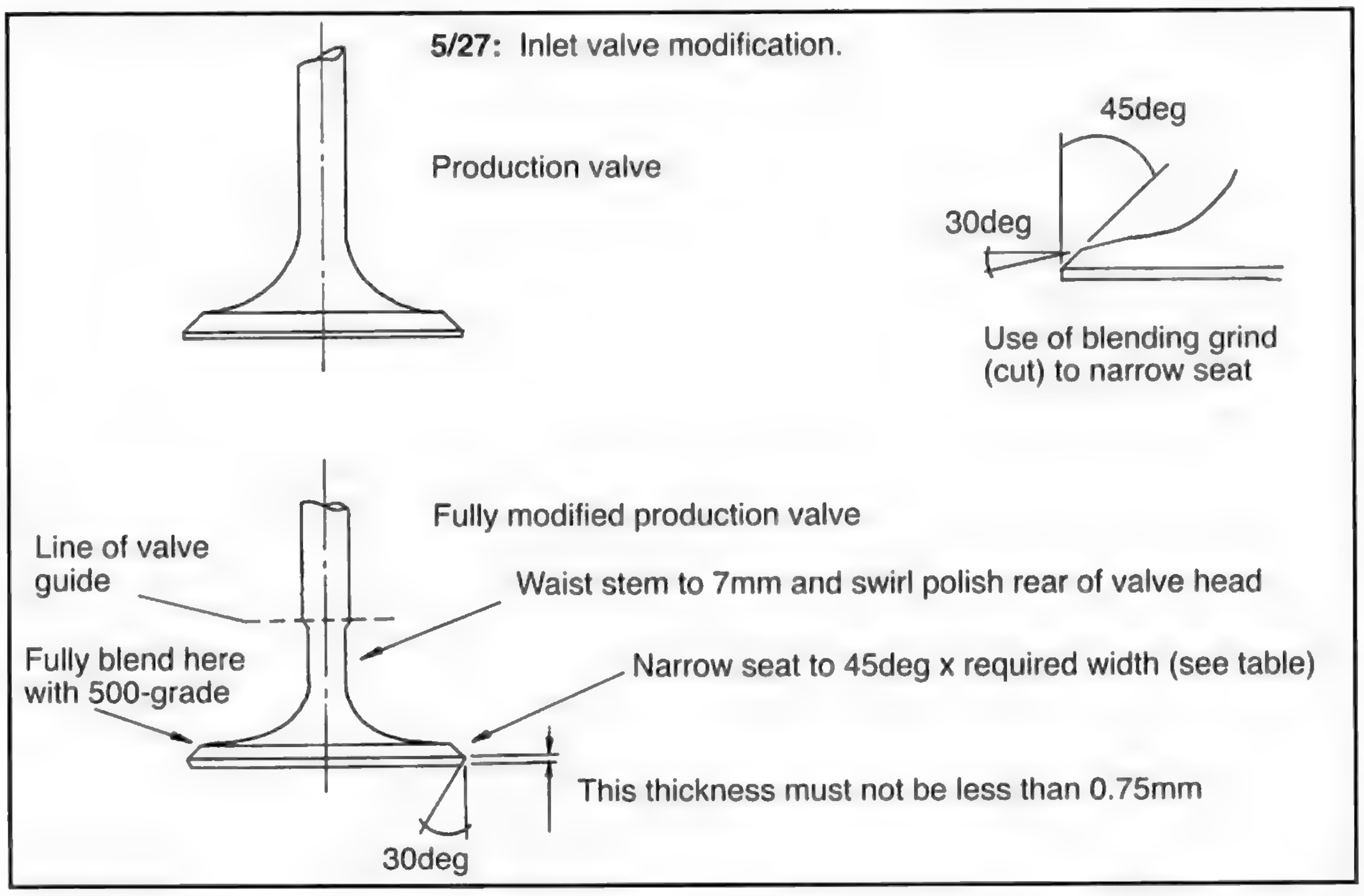
The flow characteristic of the standard TC valves is very good owing to their relatively slim design, but simple modifications can be made to them to enhance their performance.

The reduction on the back of the valve and stem waisting may be carried out readily using a pillar drill and Garryson Spiraband or ATA Ataband; this technique will also produce a swirl polished finish, which improves swirl as the fuel/air mixture enters the combustion chamber. (Do this after the 30deg blending cut.)

Initial removal should be carried out with 80 grit, followed by 120 grit for the final polish.

Because of the risk of the Spiraband intruding on the valve contact face, it is best to carry out the reduction work prior to grinding the contact face. The valve can then be finally blended as shown. A flowrate increase of around 12% over the standard valve alone can be expected.

It is advisable not to waist the stem inside the guide in order to maintain its support of the valve stem and keep it concentric. As a quicker alternative to the full process above, the standard valve may be given a 30deg cut as shown (**5/27**). (*Note:* For swirl polish: drill speed 400rpm, grinder 8000rpm max.)



5/27: Inlet valve modification.

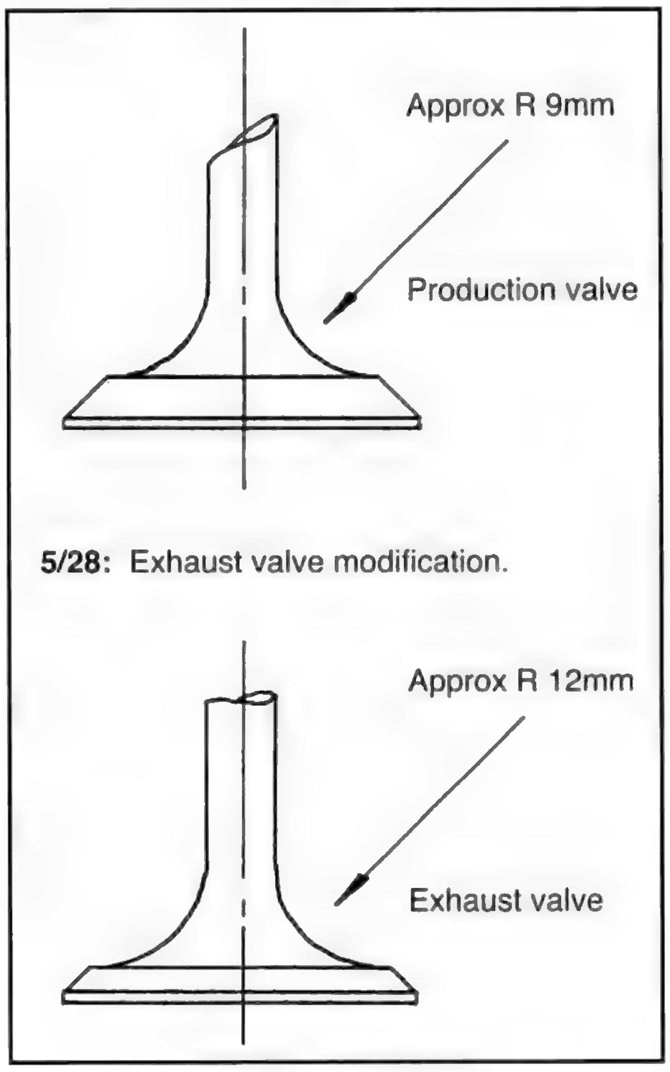
Porting

Porting (gasflowing) is the generic name for modifications to the ports to improve the flowrate/velocity characteristics, thus leading to improved cylinder filling (volumetric efficiency).

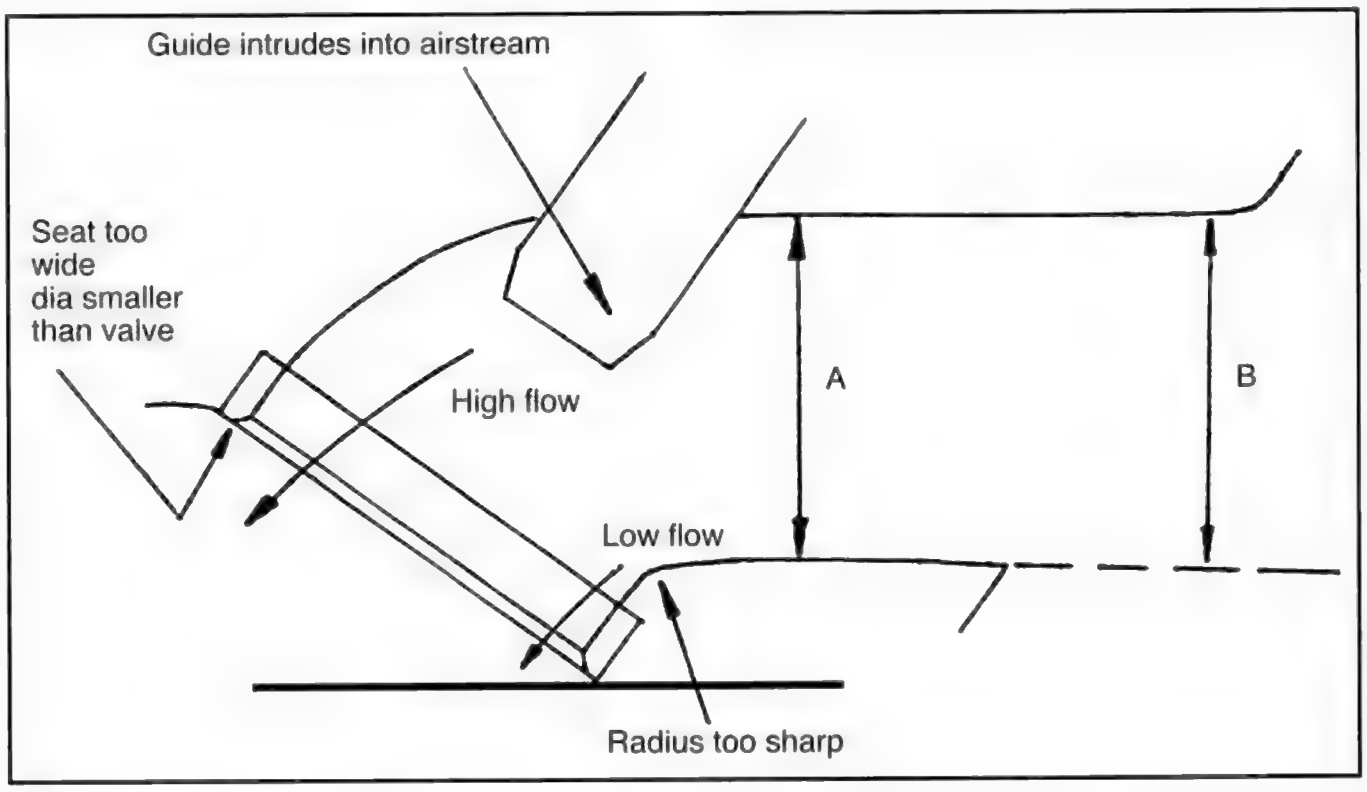
Porting should be carried out after the seats have been cut and before the head is refaced (to avoid damage). Do not shorten the guides (GCT's first job in porting) until the seats have been fully cut, blended, lapped and blued (and valves numbered) because the shape of the shortened guide can upset the position of the valve seat cutter pilot. Give the valves a final check (lap/blue) after all porting is complete.

B – Exhausts

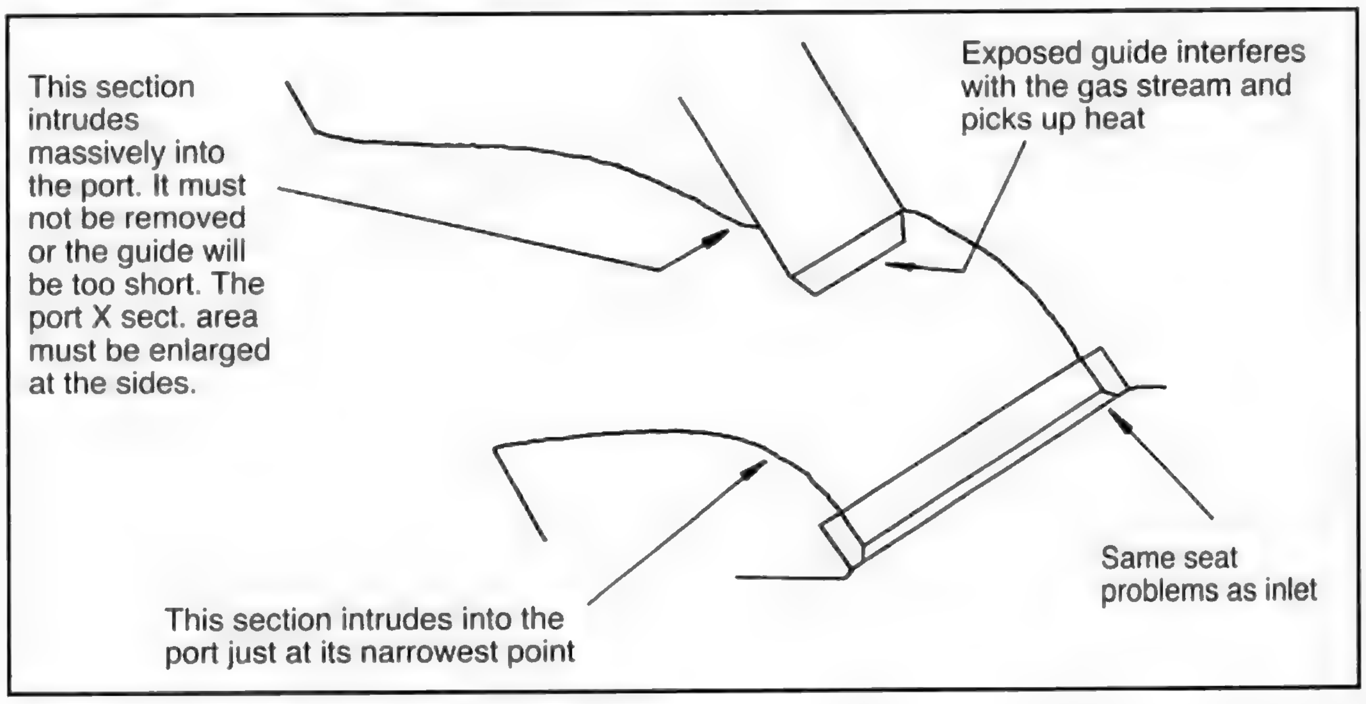
Exhaust valves may be reduced on the back of the valve and the seat width reduced (to the same width as the seat insert). The stem must not be waisted because it will upset the heat transfer to the guide, and the valve head should not be given the 30deg anti-reversionary cut. If a Spiraband (or similar) is used to narrow down the back of the exhaust valve a swirl finish will result, which will not enhance the performance of the valve, but neither will it impair it. The reduction work may also be carried out using a lathe and a radiused cutting tool (5/28).



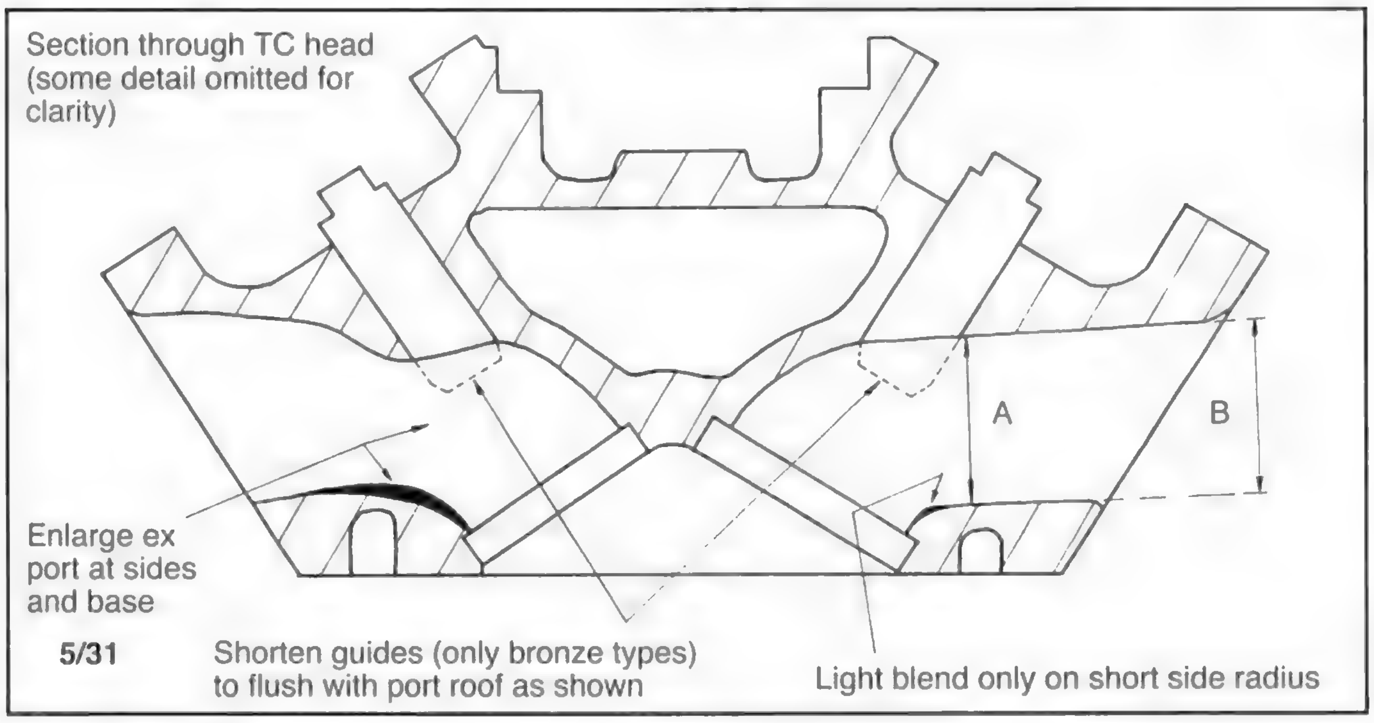
5/28: Exhaust valve modification.



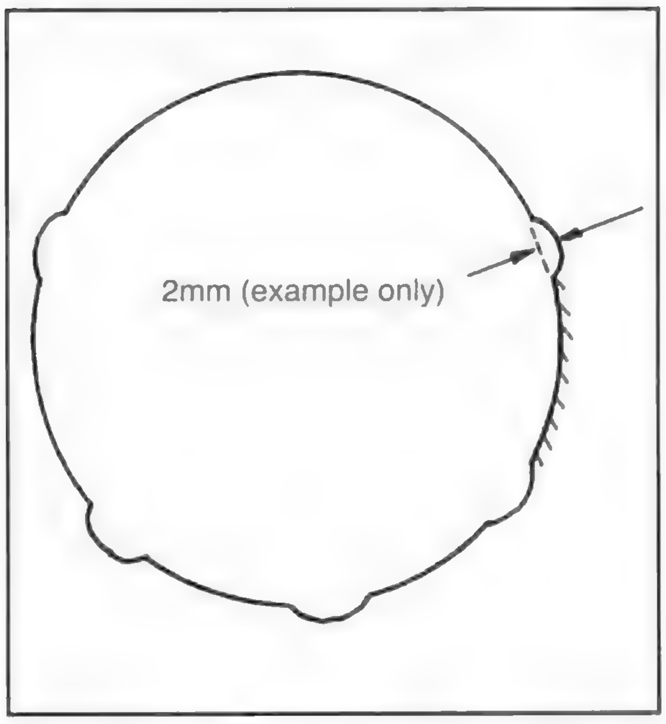
5/29: Problems with standard inlet port. Inclined valve layout gives fair degree of downdraught to port, resulting in above-average flow around 'short-side radius' adjacent to valve throat. Modifying this area of port is a trade-off between increasing port diameter at A to increase mass airflow, thus reducing this radius, which ideally should be as large as possible.



5/30: Problems with standard exhaust port.



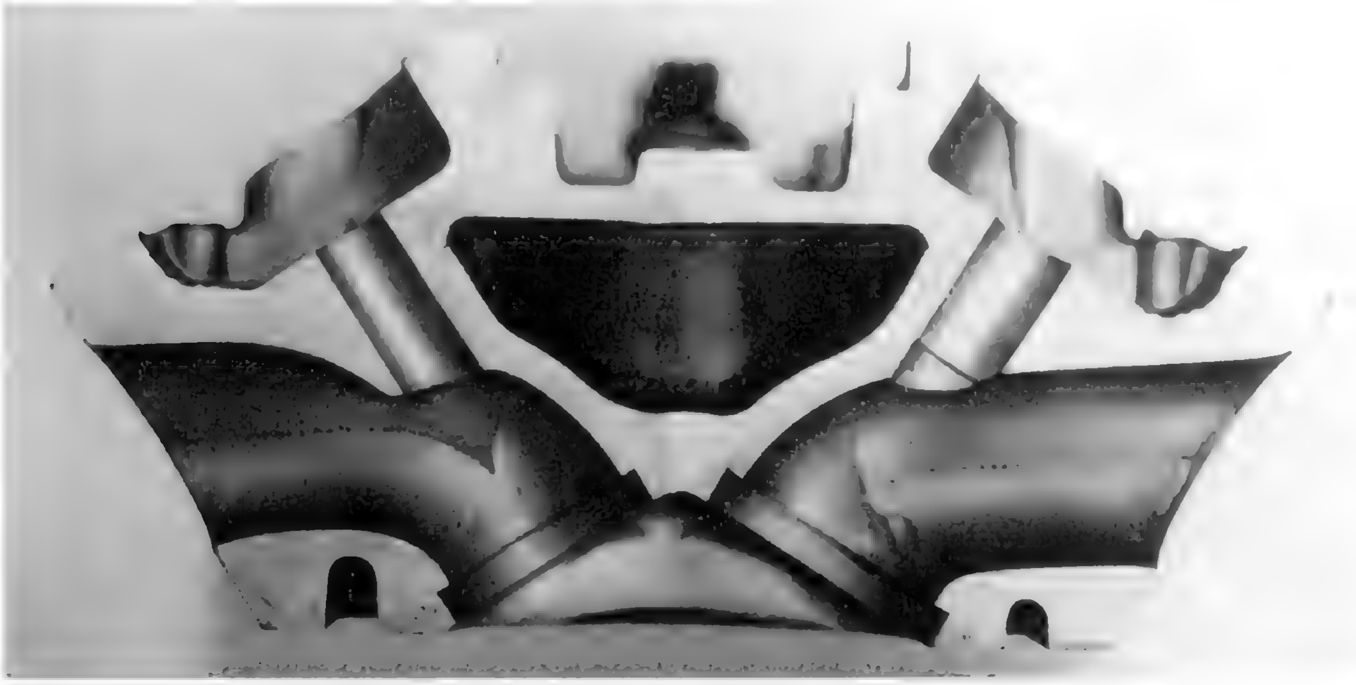
5/31



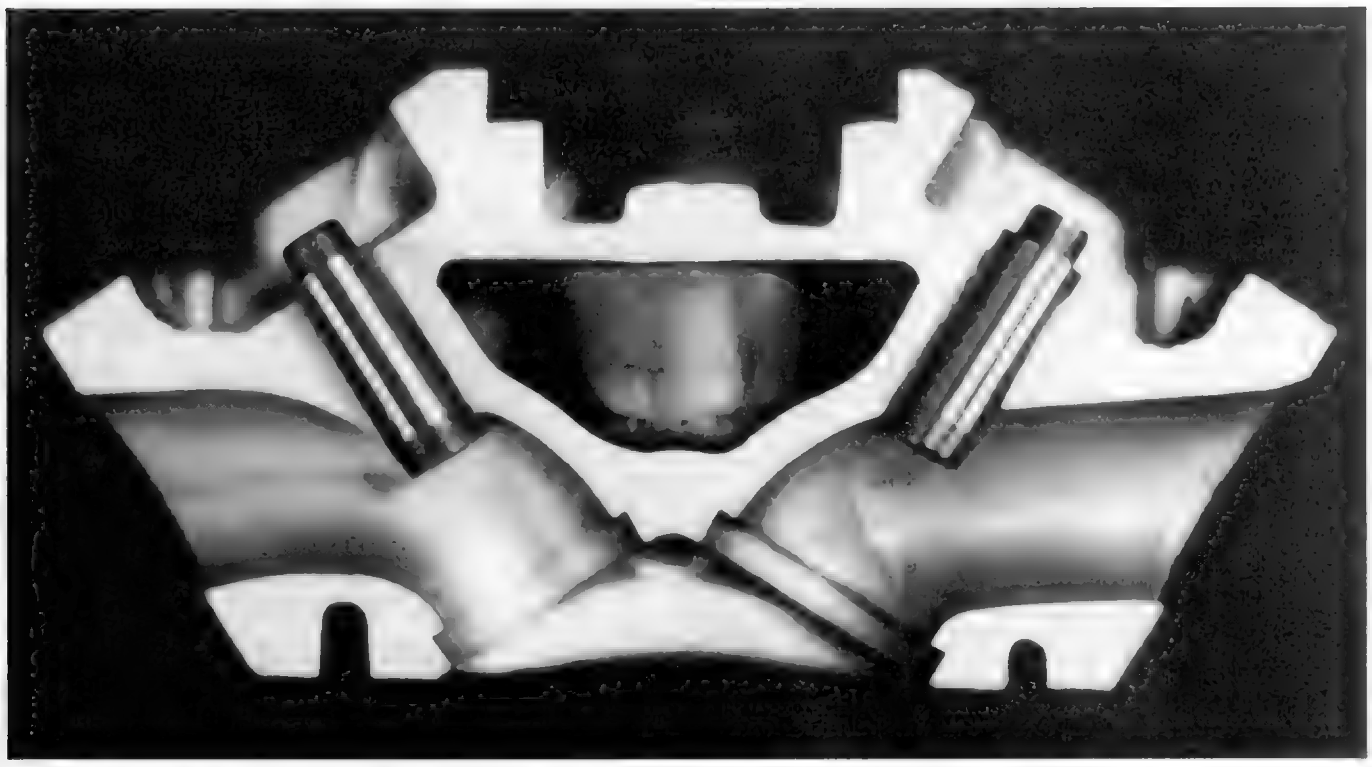
Practical porting: Exhaust port must not be enlarged at port roof in area of guide, or guide will be too short. Because of intrusion of alloy supporting guide, metal must come off base and walls of port as indicated. Dimensions A and B shown are for 131/132 2l and are nominally 32/36mm respectively. On 2l types A can be increased usefully to 34mm with all cam types by removal from port walls and base, but keep short-side radius as large as possible. On other engines dia A will depend on carb/cam set-up. Dimension B should be 36mm for all 2l types up to St III (180bhp-plus), where it should be increased to give progressive taper from carb butterfly down to A.

Roughing-out of ports can best be achieved with 'egg-shaped' burr, moving to Ataband for smoothing and finishing work. All GCT dyno results have been with a (wet) 80 grit finish; going finer seems to yield no added torque and certainly anything finer than 220-grade will lead to loss of airflow as laminar sub-layer will break away from inlet port wall. Port roof in/ex is only enlarged when a big-valve conversion has been undertaken.

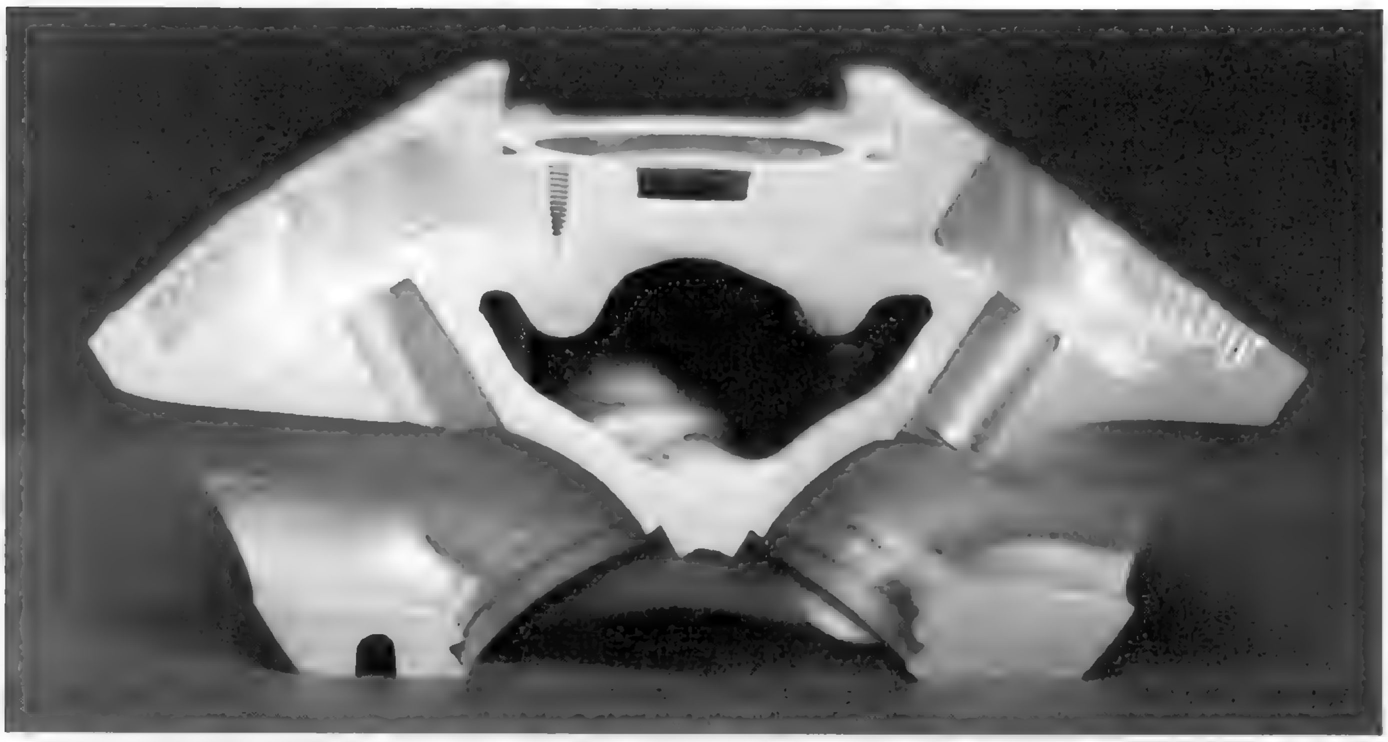
5/32: To achieve consistent increase in port dia all around, it helps to cut small channels with burr as shown and then remove hatched area to same depth. Final dia is then assessed with calliper.



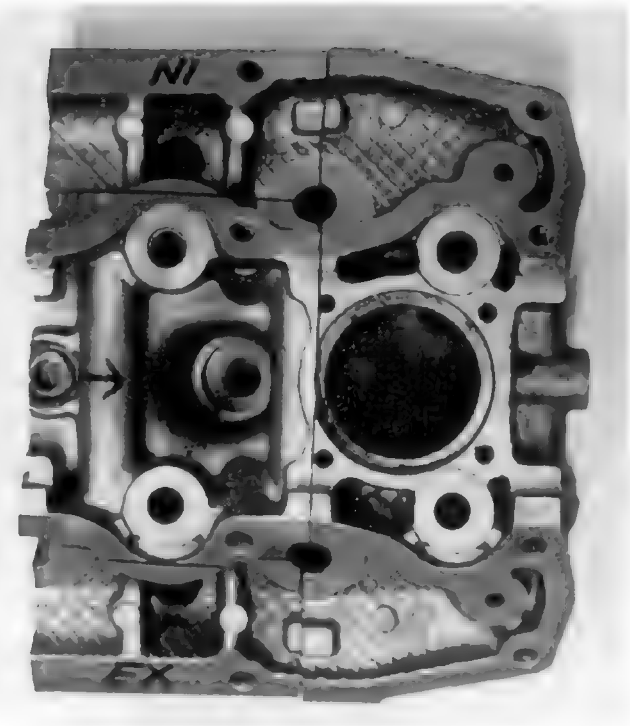
5/33: Looking to rear of engine: section across 131 2l head (inlet RHS). On big-valve large-port heads, section shows allowable space for metal removal. Parting line of casting (in two parts) is visible in ports. Crossflow effect created by valve areas presented across combustion chamber in this design gives excellent cylinder scavenging of residuals and is one reason for excellent torque. (Seat inserts have been removed.)



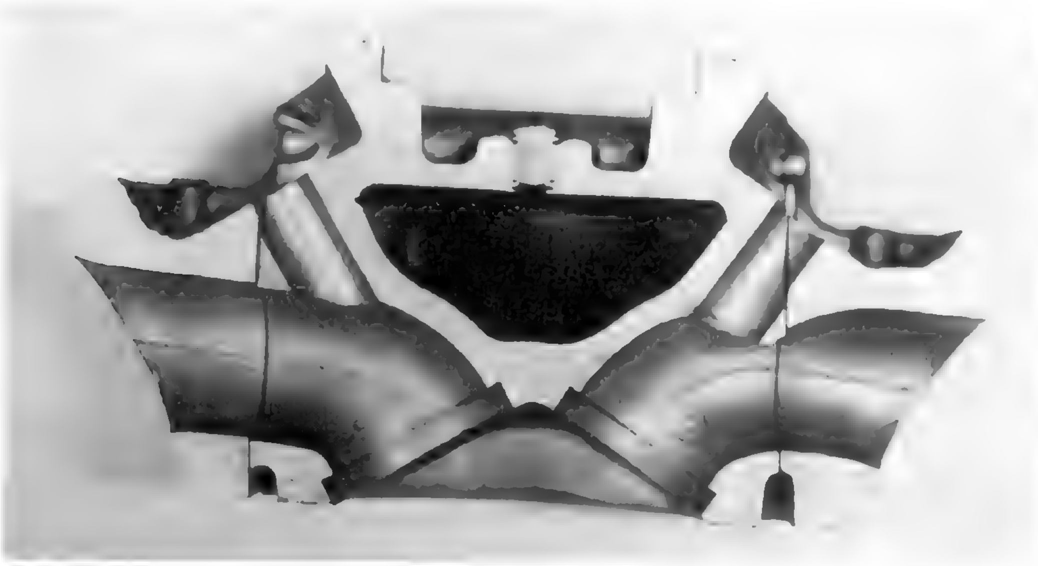
5/34: Showing how valve guides protrude into ports and upset gasflow. Note limited clearance to coolant galleries at base of port. Galleries should be welded-up first on big-port engines.



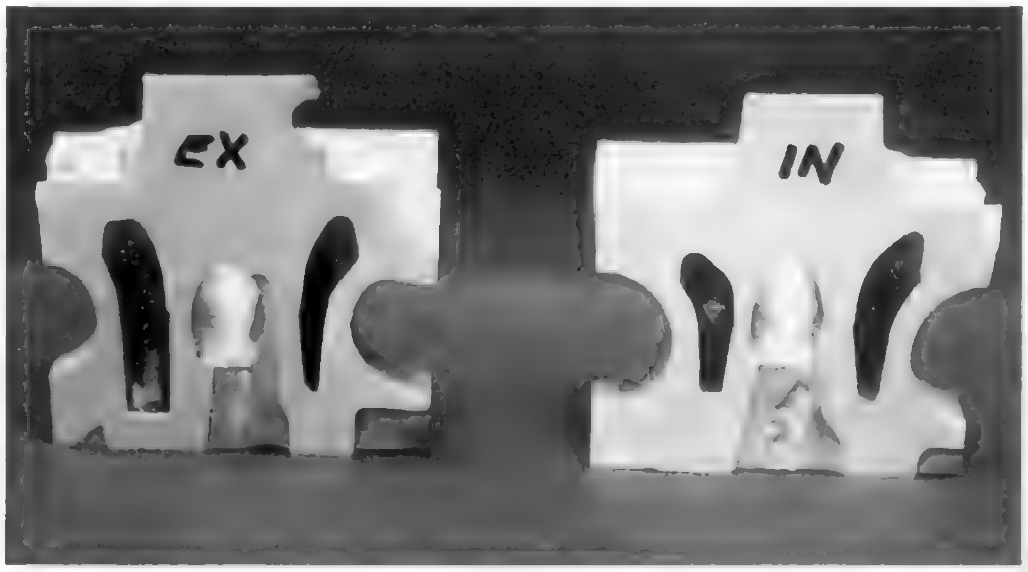
5/35: Front section of head; 8v head design hardly changed over 20 years – a tribute to success of original layout (and quality of patterns used for casting). Why change an excellent design?



5/36: Shots of front head section of 131 2l showing sections cut for following photos.



5/37: Looking towards front of head, inlet on left.



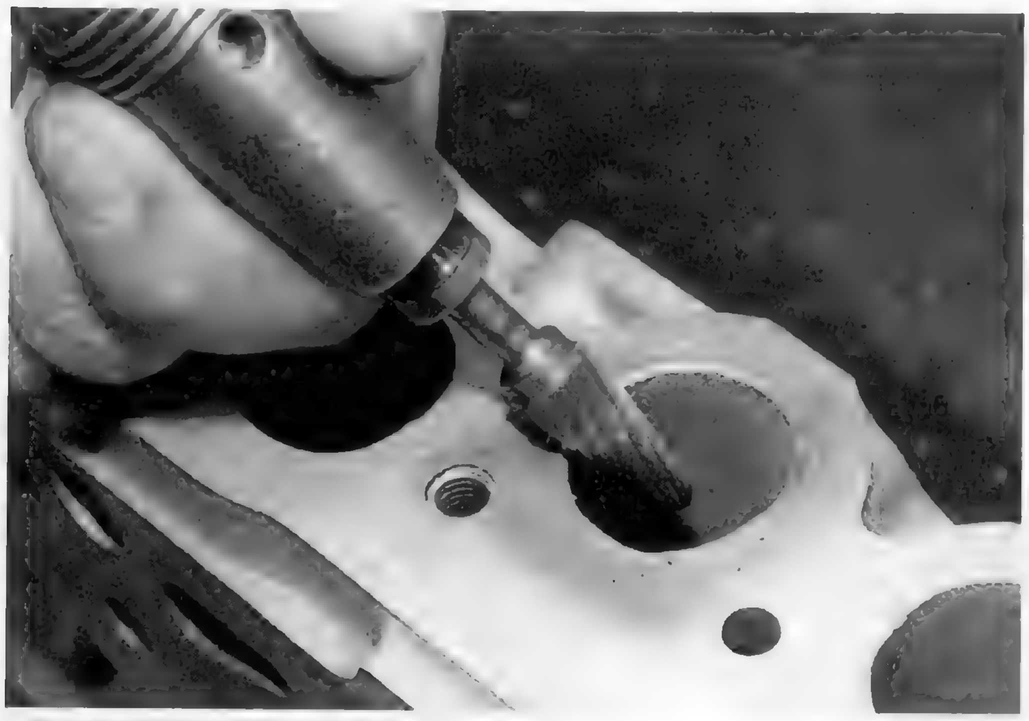
5/38: Section along crank axis on inlet and exhaust ports shows amount of metal available for removal for enlargement of ports. A minimum of 3mm thickness should be retained.

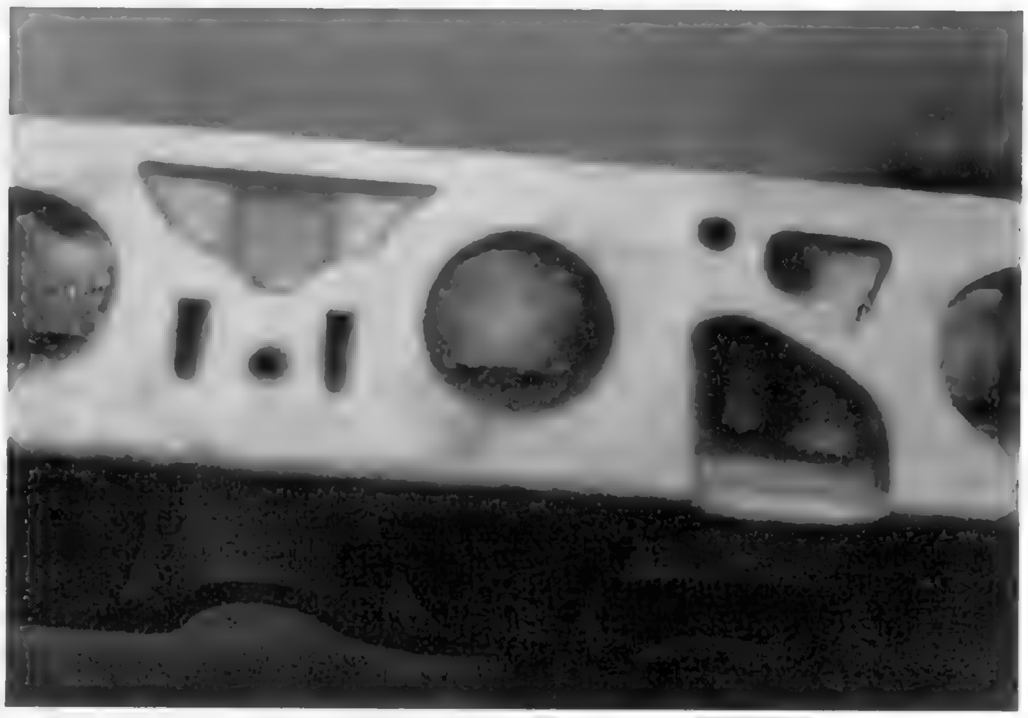
5/39: Refacing big-valve head using mill and single-point carbide cutter. Note welded-up coolant galleries on inlet side. Tool dia is 10", speed is 570rpm @ 120mm/min. Considerable amount of blending can be carried out on inlet and exhaust ports with all valve sizes at area arrowed. If machining head or block to raise CR, ensure radial and vertical valve-piston clearances are not reduced dangerously (see Chapter 14 – Dry-building).

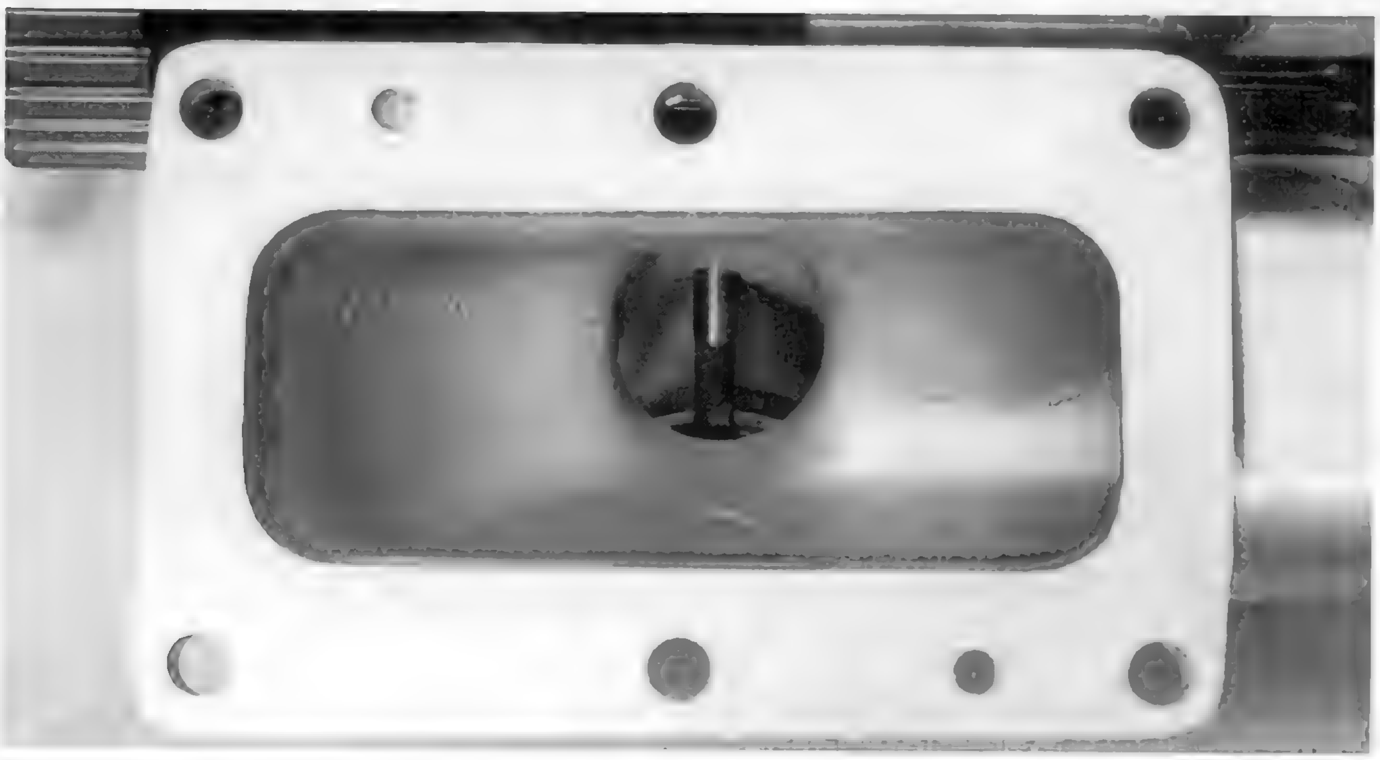
5/40: Porting tools used most frequently. Heavy-duty ATA air grinder (top) capable of 8000rpm. High torque output is good for roughing out ports and reducing valve guides. Lower tool is Metabo flexi-shaft. Coupled to Metabo electric die grinder, speed is controlled via transformer up to 27,000rpm. Most work in alloy uses around 7–12,000rpm. Lubrication of work area with spray oil (eg WD40) helps prevent tools from clogging.

Burrs shown are tungsten carbide. ATA and Garryson range will cope with anything you are likely to encounter. High-speed steel burrs only have about 1/10 working life of carbide. Although carbide ones can cost £15–£30 each, they can be resharpened for 1/3–1/2 cost of new. ATA Atabands accept various grades of carborundum bands for blending work. Stones can be used, but lubricate with paraffin or WD40.

5/41: If matching of gasket or manifold to head is required, scribe as shown or spray through manifold/gasket with aerosol paint. If port is to be opened out, start with area around this mark first. This will give a diameter to work to as you move deeper into port.

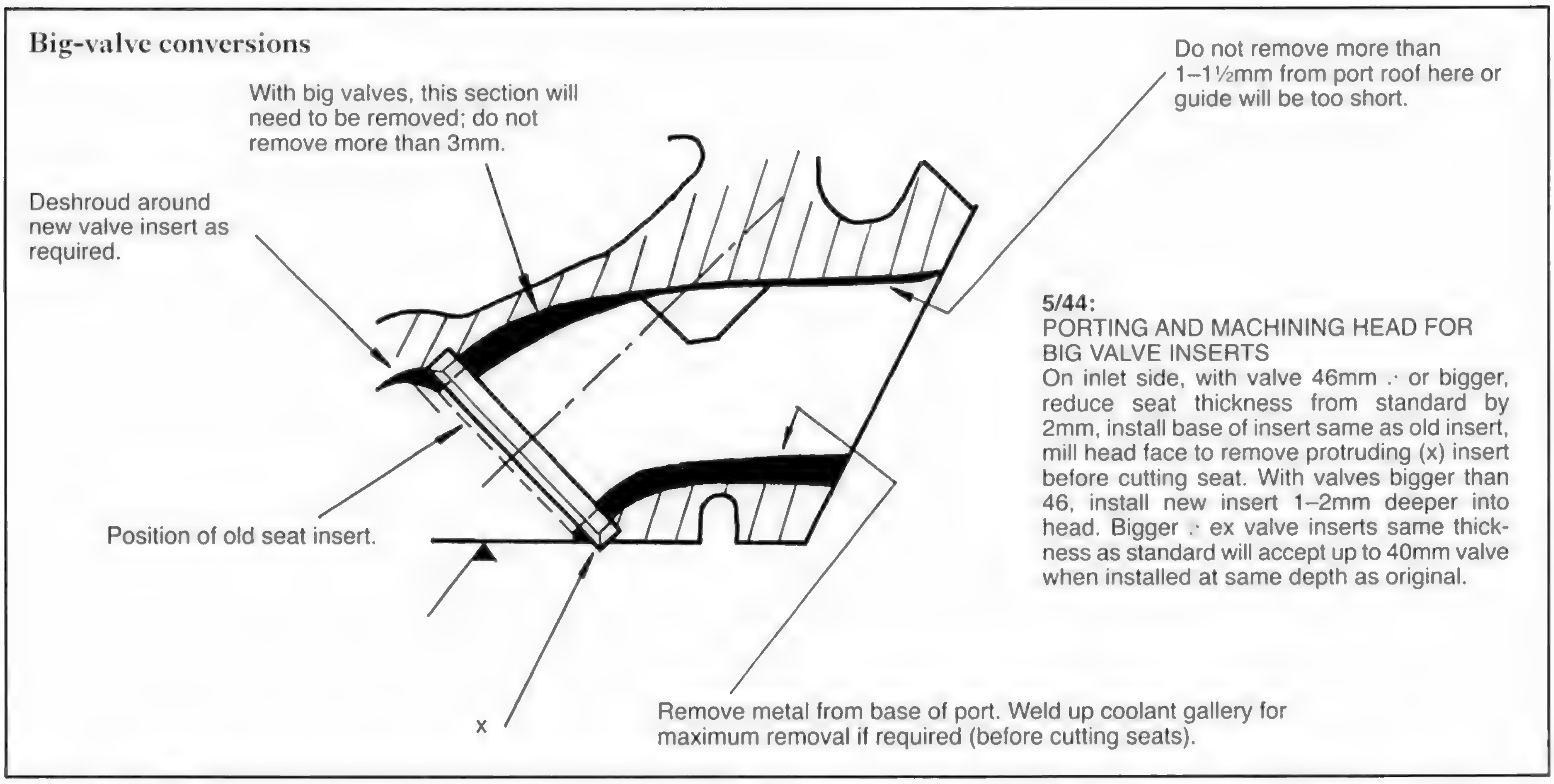




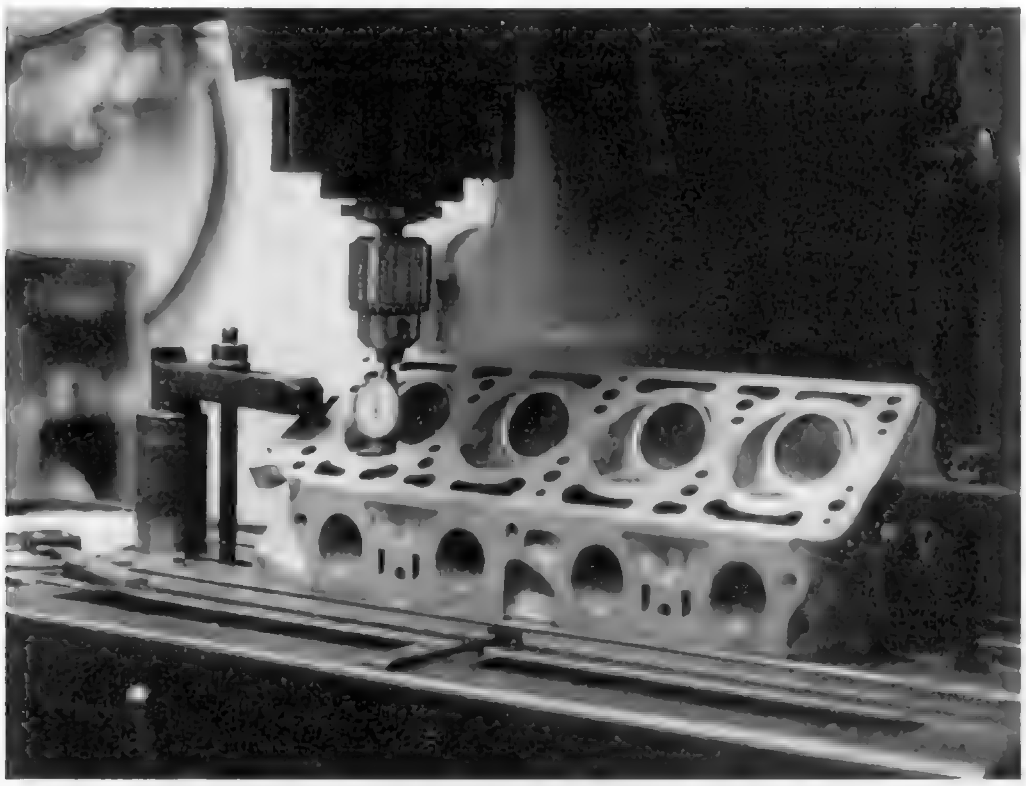


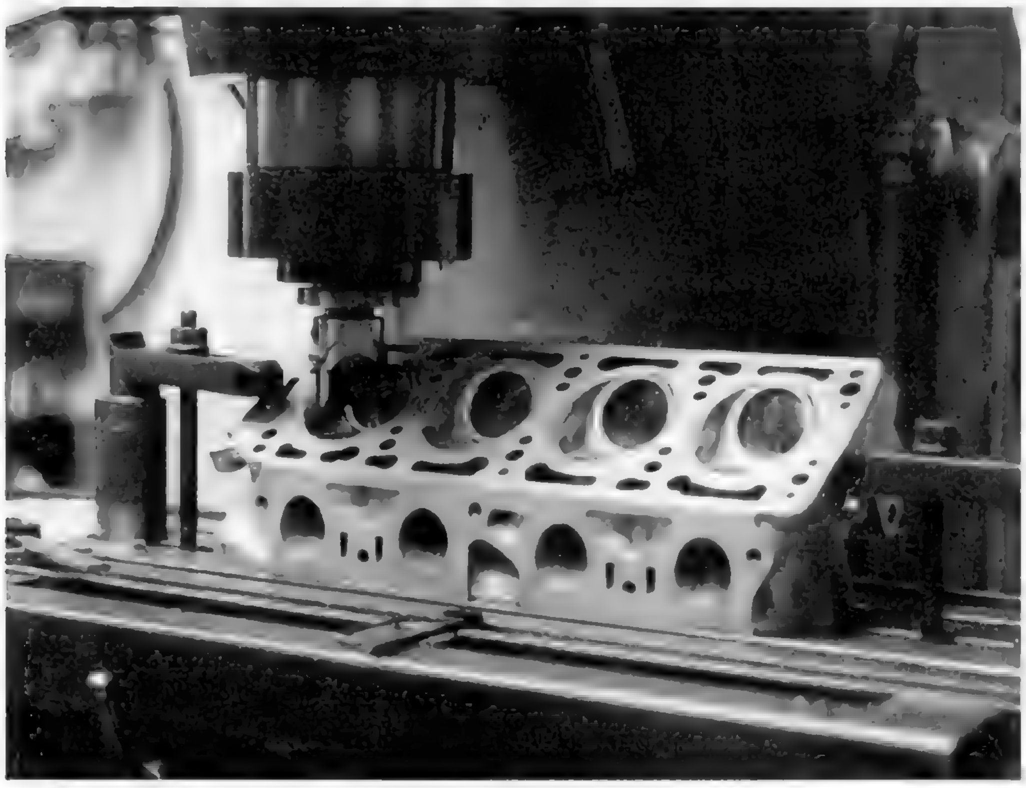
5/42: View of exhaust port with guide shortened. Ports have been roughed out using carbide burr shown and are ready for finishing with Ataband. Note that exhaust guide has only been shortened back to alloy section, no deeper. Exposed exhaust guide only picks up heat from exhaust gas.

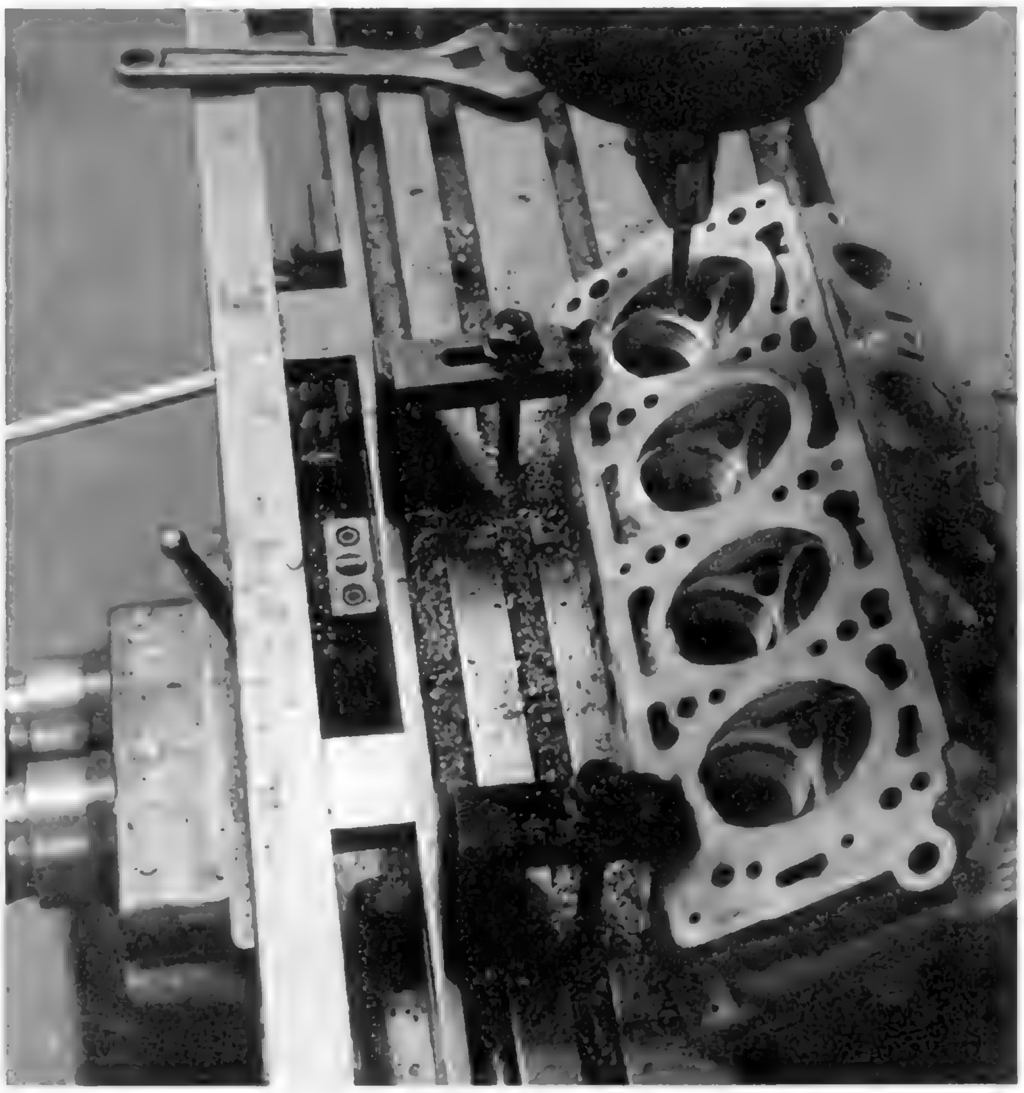
5/43: Finished inlet port of Lancia Volumex showing guide cut off at port roof. Very important to use bronze guides if they are going to be shortened – cast iron guides can crack under small lateral shift caused by running clearance of cam bucket in its bore. It is not necessary to use standard-length tapered guides: with short protruding valve stem, shortened guide has plenty of residual strength to keep valve centred. (This is an irrefutable fact based on long-term tests of dozens of GCT competition engines.)



5/44:
PORTING AND MACHINING HEAD FOR BIG VALVE INSERTS
On inlet side, with valve 46mm .· or bigger, reduce seat thickness from standard by 2mm, install base of insert same as old insert, mill head face to remove protruding (x) insert before cutting seat. With valves bigger than 46, install new insert 1–2mm deeper into head. Bigger .· ex valve inserts same thickness as standard will accept up to 40mm valve when installed at same depth as original.







5/45, 5/46, 5/47:
BIG VALVES
Big-valve head underway on mill. Single-point carbide tool has been used to bore out head to accept larger inserts. Inserts (alloy bronze inlet) pressed in, then cutter used to enlarge throat area to save time. Tool is centred on new guide using long finger dial gauge. Interference fit is critical. Mill can be used to press seat inserts into head; no heating of head is needed, indeed doing this cold gives good idea of tightness of fit.
Valve insert fitting (GC recommended fits):
Alloy bronze inserts – 45–50mm dia, 4–5thou" interference
Nickel steel/cast iron inserts – 37–42mm dia, 4–6thou"
If a seat is too loose, there is no option but to fit a larger one!

Part 3: MEASURING COMPRESSION RATIO

CR is given by the formula:

$$CR = \frac{Vs + Vc}{Vc}$$

or in other words:

$$CR = 1 + \frac{Vs}{Vc}$$

As described in *Chapter 2*, compression ratio has a major influence on the thermal efficiency – and hence power output from a given amount of fuel (**5/48**).

Vs = swept volume of one cylinder calculated as follows:

$$Vs\ (cc) = \pi \times \frac{D_b^2}{4000} \times S$$

where D_b = bore size (mm)
S = piston stroke (mm)

Vc = clearance volume above piston at TDC calculated as follows:
Vc = a + combustion chamber volume
b + volume of valve cutouts in pistons
c + compressed gasket volume
d + volume of bowl in piston (if applicable)
e + volume between piston crown and block face if low in block at TDC (top clearance)
f – volume between piston crown and block face at TDC if piston above block (top clearance)
g – volume of piston dome (if applicable)

Items f and g are 'intruder' items and will increase CR. Other items tend to reduce the CR and should be made as small as possible. Items c and e can be calculated using the same equation as Vs, *ie*:

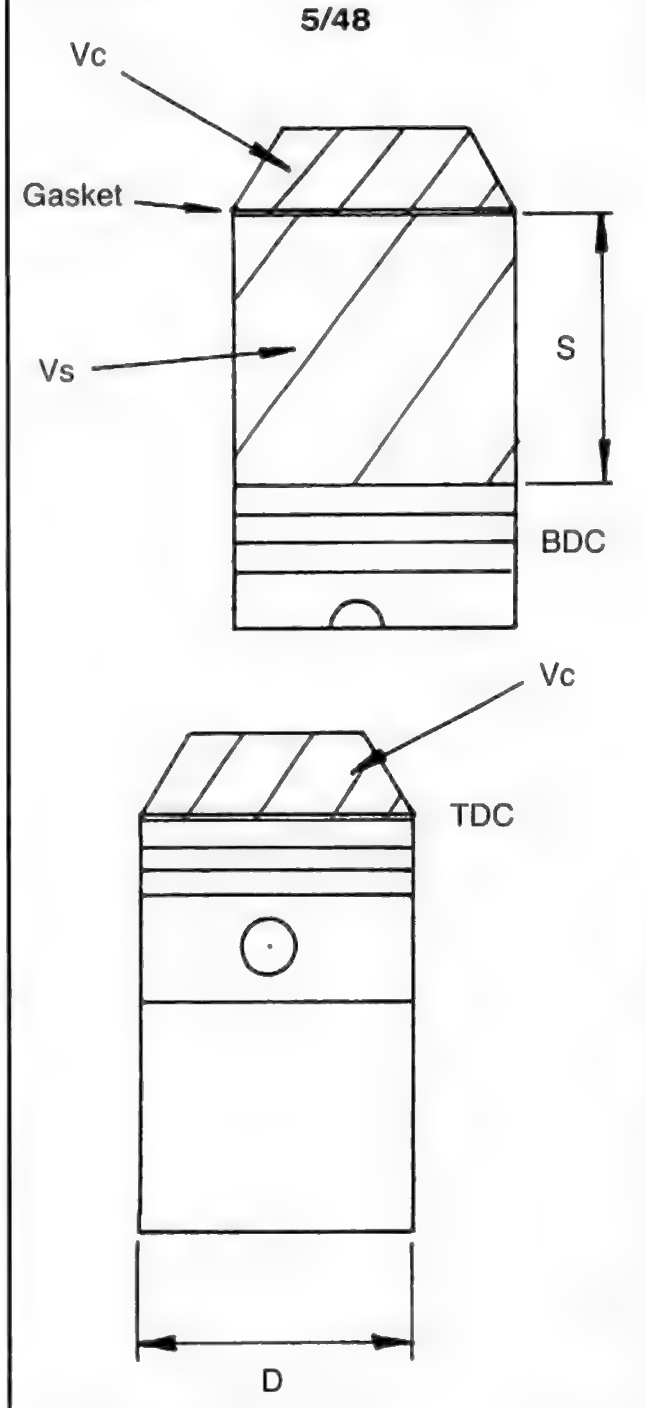
gasket volume (cc)

$$= \pi \times \frac{(\text{dia (mm) of gasket bore})^2}{4000} \times \text{gasket compressed thickness (mm)}$$

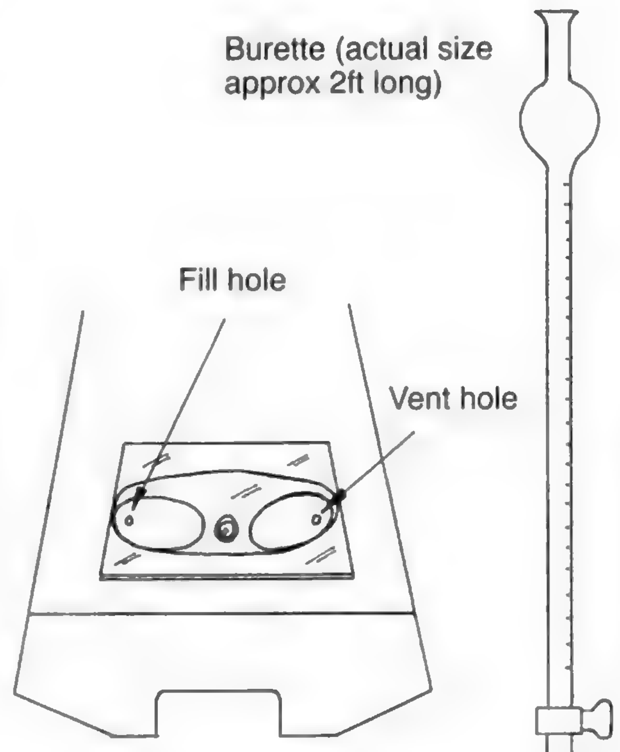
top clearance volume (cc)

$$= \pi \times \frac{(\text{dia (mm) piston})^2}{4000} \times \text{height of crown above or below block (mm)}$$

5/48



5/49



Perspex sheet in position over combustion chamber

Buretting

Items a, b, d and g must be measured with a burette. This glass, graduated measuring device is filled with paraffin (or similar) and used to measure the appropriate volume with fluid (**5/49**).

Combustion chamber volume (head volume)

Grease the valves in the chamber to be measured, screw in the spark plug to be used, fit the valves (ensuring the grease is evenly distributed so no fluid leaks into the ports), then grease around the chamber and lay a small Perspex sheet over the chamber as shown. (Perspex sheet may be purchased from a glazing shop.) The sheet should have two holes in it as shown and the head should be set at a very slight angle to allow the air to escape out of the vent hole. Bleed the fluid into the chamber and measure the amount in (cc); (a measurement difference between chambers of ±0.5cc is perfectly acceptable if they are being checked for size).

During buretting, ensure the ports remain dry; any leakage and the result will be meaningless! If the inlet valves protrude beyond the head face, grind a small relief into the Perspex plate. Measure the volume of fluid when the combustion chamber is completely filled.

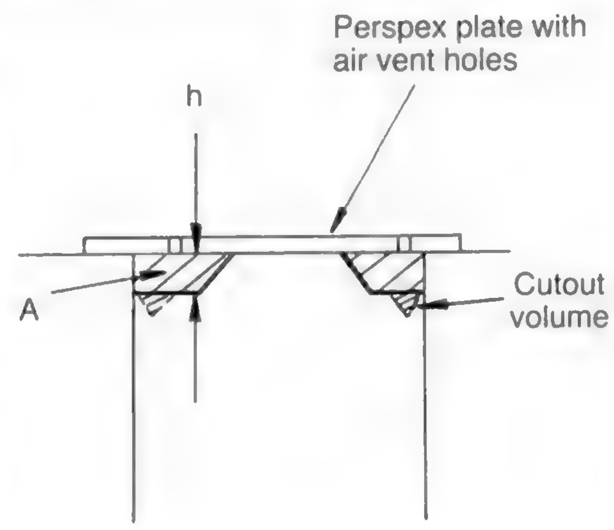
Piston dome volume, bowl and cutouts
Domed pistons

In the case of domed pistons the cutout volume may be measured with acceptable accuracy by simply laying the piston on a flat surface and filling the cutouts. To measure the volume of the dome, place a piston in a bore (with rings fitted) and grease around the side of the piston to seal it in the bore; the top of the dome should be flush with the top of the block. Measure height 'h' *(see diagram)* with a calliper, then grease around the edges of the bore and place the Perspex plate over it. Fill the space 'A' with fluid from the burette (**5/50**).

The volume of the dome is calculated as follows:

$$\text{dome volume} = \frac{\pi D^2}{4000} \times h \text{ (mm)} - [\text{vol (cc) of fluid to fill A minus cutout volume}]$$

5/50: Buretting piston

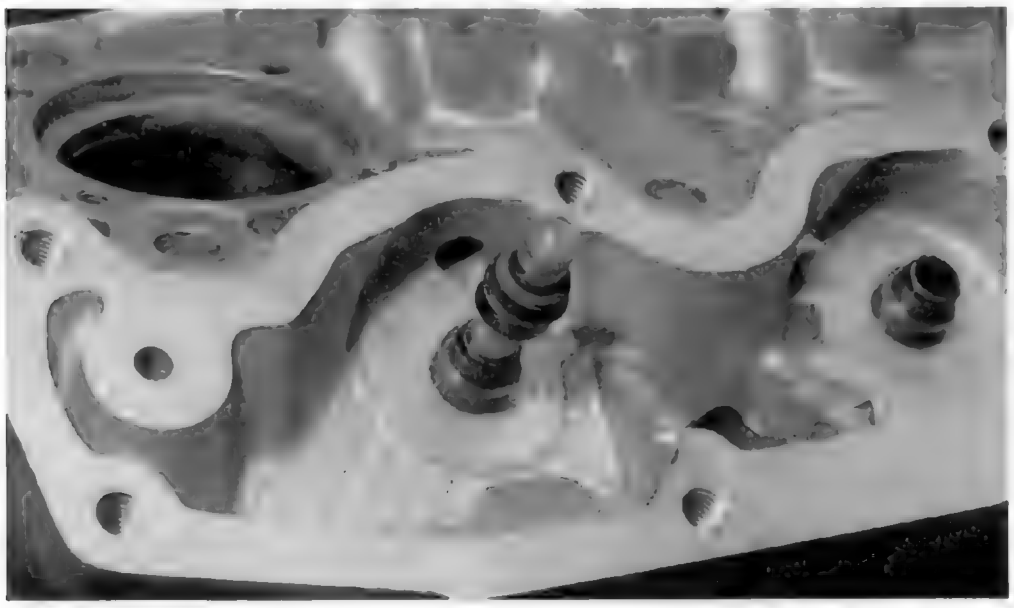


Bowl-type pistons (*eg* turbo, Volumex)

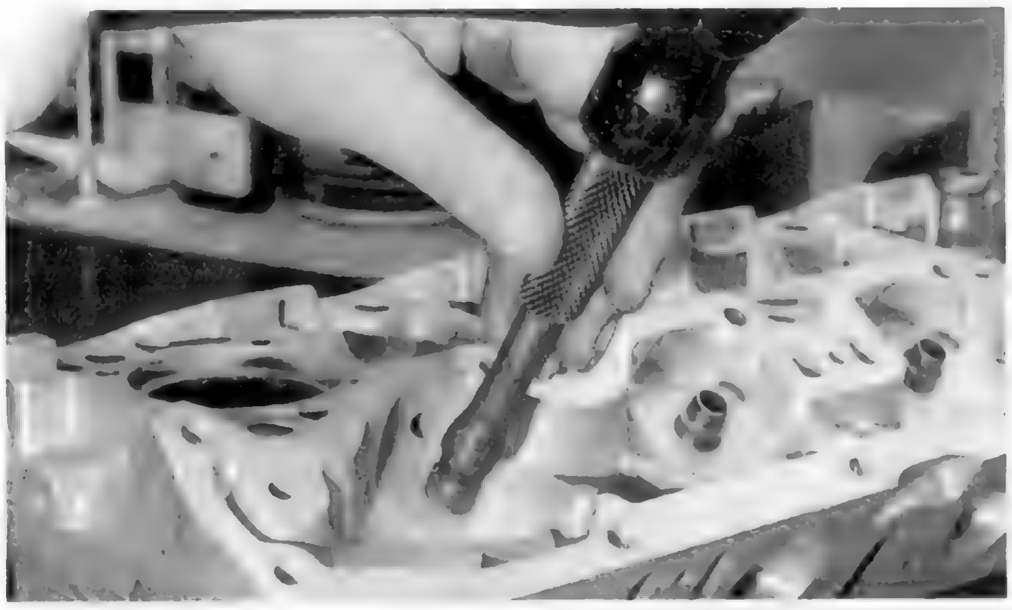
In this case, the bowl and cutouts can be measured together by placing the Perspex sheet over the top of the piston and filling the bowl and cutouts together.

BUILDING UP THE HEAD

Before starting, is a dry-build required? *(see Chapter 14)*

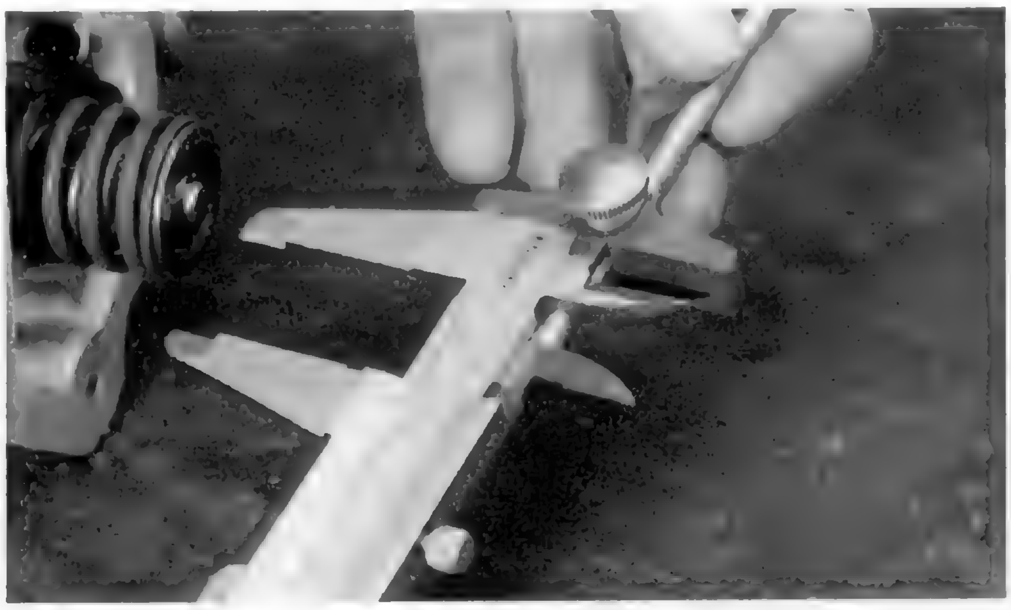


5/51: 1 – Lubricate valve seats and stems with oil or graphite grease, fit spring seats and fit protective sleeve over valve tip; this protects stem seal from damage from collet groove (this can halve stem seal life). Stem seal shown (8v) is AE (Payen) part no HR 269, which is stronger than early-pattern seals. Slip seal over sleeve and…

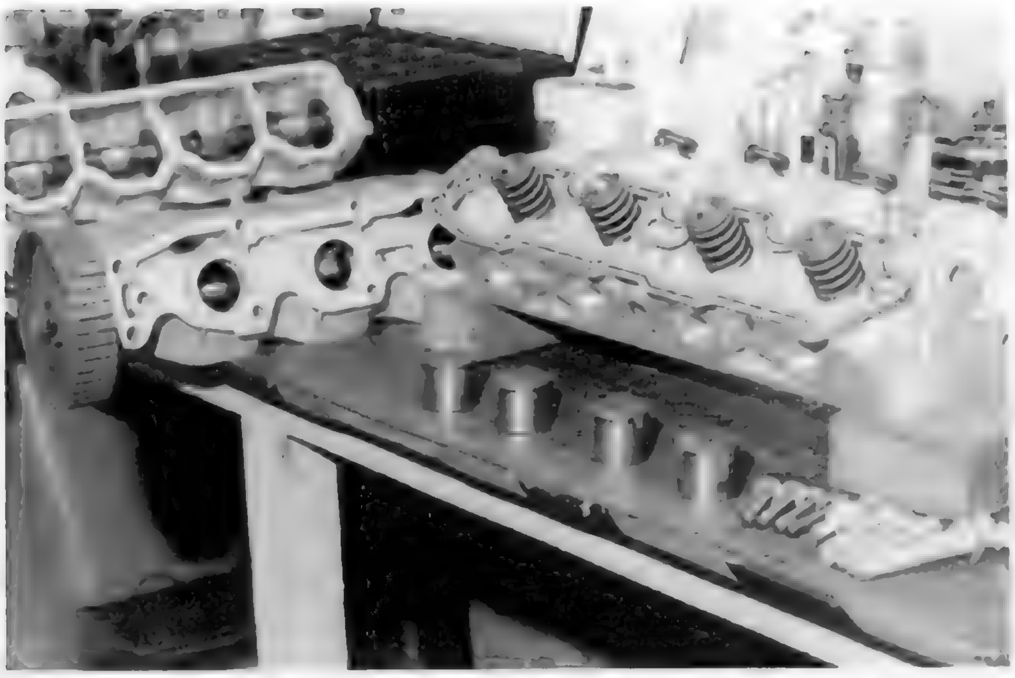


5/52: 2 – …tap seal into place. Seal should hold firmly on valve guide. If you don't have drift to do this, a 12mm deep drive socket works well enough. [Author's note: I forgot to fit the spring seats first on this head – and couldn't get them on!]

5/53: 3 – Fit valve and springs (using spring compressor), tap valve tips lightly to ensure collets are correctly seated. To aid shimming if seats/valves have been recut, measure protrusion of valve tip above mating face of head above cam box. (This can be done before fitting seals and springs – in case valves need to be removed and 'topped', ie stems shortened in valve refacing machine.)

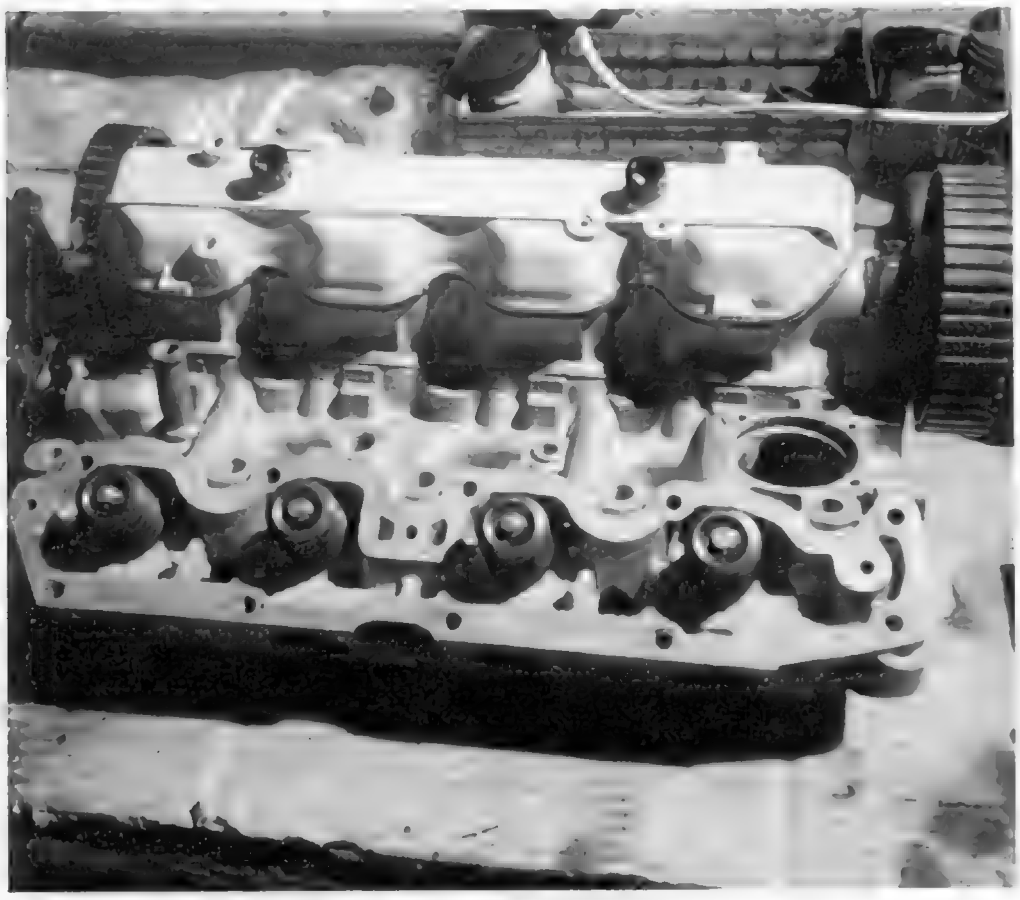


5/54: 4 – Standard distance is 22mm; when inlet/exhaust seats are blueprinted, valves drop deeper into head, so distance will be 22.5mm or greater. Advantage of doing this is that shim thickness can be estimated before head is fully built-up. Distance from cam box base to cam base circle can also be measured; standard cams give approx 29.2mm. Hence (since Fiat tend to use standard shims around 4.20mm thick), if valve height is 23mm (+1) and base circle distance is 30.2 (+1, due to reduced base circle on a comp cam), standard shim thickness of around 4.20 is going to be about right with one cam box gasket in place. It helps to use a shim thickness perhaps 0.2mm less, just to make sure shims are not too tight to get a feeler gauge in to measure clearance.

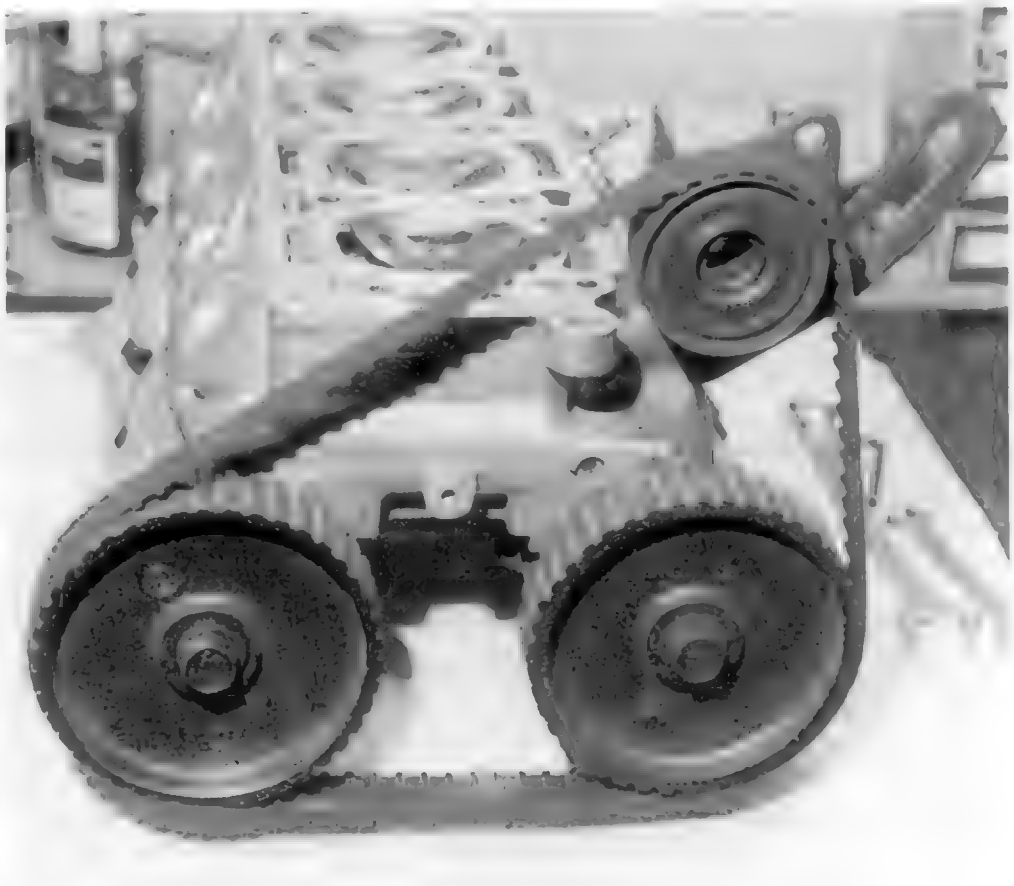


5/55: 5 – Head in this case needed 2 cam box gaskets: base circle distance was 29.2 (std 2l Beta), ex valves protruded 23mm above cam box mating face on head. Reason for 2 gaskets? Exhaust valves were +1mm up – standard shim of 4.20 would have meant starting point for shimming without two gaskets would have been 3.20 – too low to allow for any further reduction. Valve tips can also be ground down, but if engine is later tuned-up with competition cams (reduced base circle) this increased valve height will be an advantage. (Note latest pattern AE cam box gaskets – installed dry.)

5/56: 6 – Lubricate valve tips, buckets and fit new shims (or invert old ones if unused on one side) and bolt down cam box on one side. Shim-up one side first, then check to see if there is enough clearance between valves to allow cam on other side to be rotated and shimmed without valves clashing. If in doubt, slacken-off other cam box to withdraw valves into head.



5/57: 7 – Gareth Jones' 1600 Fiat head in course of build-up with production valves and standard cams. If one cam box (in this case inlet) is set at its TDC position there is plenty of clearance between valves to allow exhaust cam box to be bolted up and shimmed. With bigger valves/competition cams – always check first.



5/58: 8 – Simple rig made from 131 alternator bracket/tensioner, bolted to head with short toothed belt, allows valve action to be assessed prior to fitting to block. This is crucial when valve sizes of 44/38 and/or cams of over 290° or 4mm LATDC are used. Minimum clearance between valves must be 1½–2mm at closest point. With valves over 45/40 it may be necessary to recut seats or alter cam timing to achieve clearance required. Reasonable clearance gives nice safety factor – a GCT-prepared Gp A Integrale 8v with Evolution inlet cam had 1mm radial 'slop' on inlet valve head at full lift after 2 seasons international rallying – due to radical profile causing extreme guide wear!

CASE HISTORY No 2

Owner	Tom Casey
Engine No	GC 222
Type	Fiat 2043cc (85mm bore)
Use	National Hot Rod
Tested	Warrior Automotive, Dec '94
Rig	Superflow

It was considered that the original engine *(see Case History No 3)* lacked acceleration out of the corners in oval racing, despite having good top-end torque, and that the spec should be re-evaluated to enhance the torque in the speed range 4000–6000rpm. Tom's engine had suffered extensive damage due to detonation caused by a faulty fuel pump and it was felt that the consequent rebuild would provide an opportunity to carry out extensive testing.

The engine was rebuilt with new pistons with a revised dome layout, raising the CR to 11.2:1, and following flowbench testing of the original head, the pattern of the inlet valve throats was modified to raise the flowrate from 117.3cfm at 12.2mm (actual) lift to 122cfm. Since the flowbench tests indicated that at 10.8mm ex (actual) lift the head flowed 95.8cfm, *ie* 78% of peak inlet flow, it was reckoned that by changing to a milder exhaust cam, with shorter duration and lower lift at TDC, the torque characteristic could be enhanced at lower revs, although some drop in peak power would probably result. The final results were to exceed all expectations! Numerous intermediate power runs were carried out to optimize jetting and ignition and the tests below represent the optimum in each case.

Test 1 results: Torque now peaked at 4500rpm, although reduced in magnitude by 6.5lbf ft, but more significantly the torque from 4000–5500 was substantially enhanced.

It was found throughout the power runs to optimize Test 1 that the engine was extremely sensitive to ignition timing. It was also found that an increase of carburettor air temperature from 17° to 24° led to a loss of 7lbf ft torque, which highlights the importance of ducting and airboxes to ensure that the carburettors receive cold (external) air rather than air at underbonnet temperatures.

It was felt that the extended duration of the GC IVA inlet cam into the compression stroke was still causing loss of mid-range torque.

Test 2 results: The torque gains at 4000 and 7000–8500rpm over Test 1 were encouraging; certainly the increase in torque of 14.8lbf ft at 5000rpm over the earlier engine *(see Case History No 3, Test 5)* was bound to lead to greatly improved acceleration out of corners. However, the torque at the top end (7000–7500) could be improved by leaning out the air corrector and, likewise, enrichment of the main jets would enhance the torque around 4500–5000. This proved to be the case.

TEST 1 GC IVA inlet cam (106°), IIIA exhaust cam

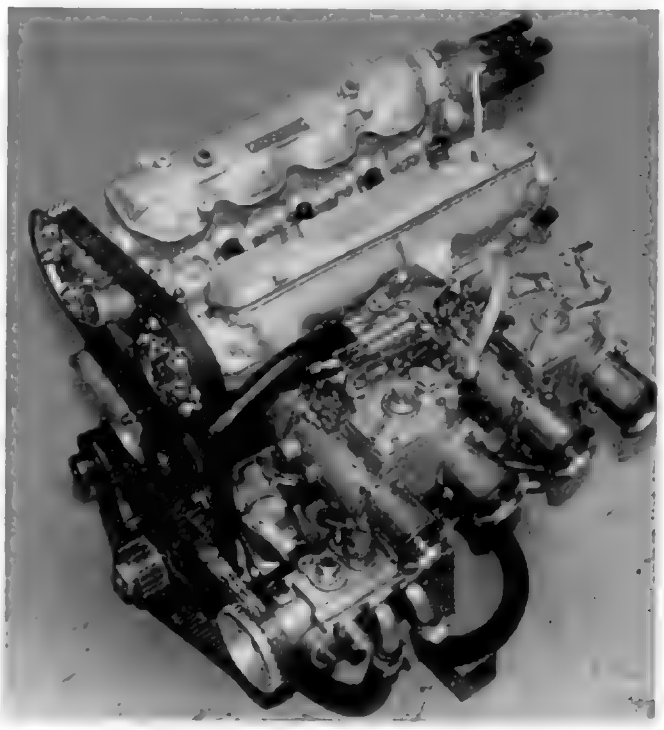
SPEED (rpm)	TORQUE (lbf ft)	POWER (bhp)	BSFC (gm/kW hr)	
4000	128.8	98.1	361	35° adv @ 5500
4500	143	122.6	308	170 main jet
5000	135.9	129.4	322	150 air corrector
5500	141.3	148	305	F16 emulsion tube
6000	142.1	162.4	297	(48 DCOE, 40 choke)
6500	137.4	170.1	288	ex manifold 4-1
7000	133.7	178.3	307	22" primary lengths,
7500	126.5	180.7	324	45mm ID
8000	121.7	185.4	326	

TEST 2 GC IIID inlet cam (100°), IIIA exhaust cam (110°)

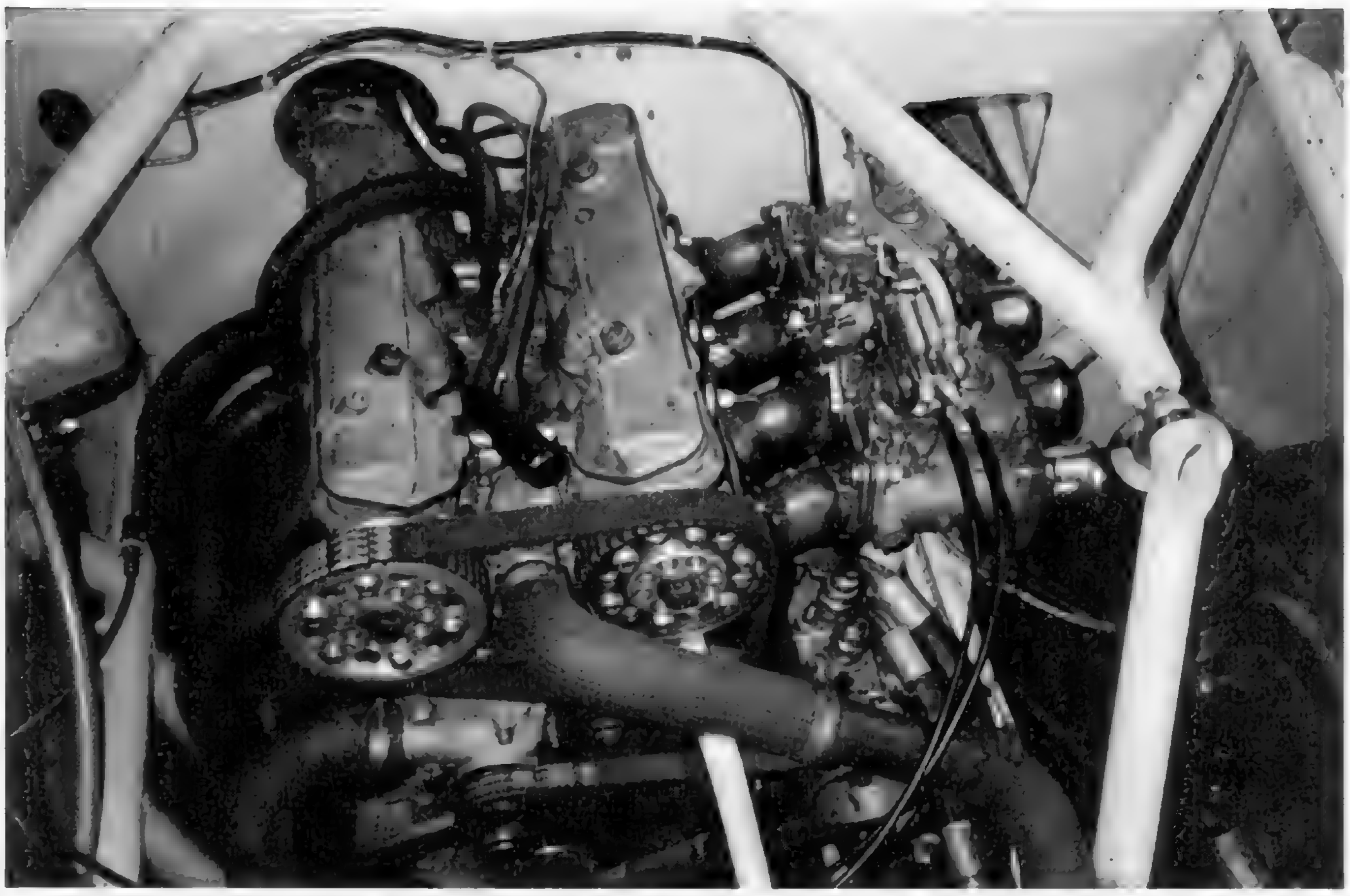
SPEED (rpm)	TORQUE (lbf ft)	POWER (bhp)	BSFC (gm/kW hr)	
4000	131.1	100	407	JETTING:
4500	140.8	120.7	318	165 main
5000	138.5	132	316	150 air corrector
5500	142.3	149.1	302	35° adv at 5500
6000	141.9	162.2	296	
6500	141.9	170	292	
7000	134.6	179.5	310	
7500	128.8	184	324	
8000	119.3	181.8	342	

TEST 3 – as Test 2 rejetted

SPEED (rpm)	TORQUE (lbf ft)	BSFC (gm/kW hr)	
4000	133.3	374	JETTING:
4500	149.9	334	170 main
5000	140.7	323	155 air corrector
5500	142.7	301	
6000	144.2	294	
6500	140.1	302	
7000	137.6	303	
7500	130.78	318	
8000	119.1	341	



CH2/1: GC engine No 222 'fully dressed' and ready for dyno-testing, features three-stage Series 2 Titan dry-sump pump and toothed-belt drive to water pump. Distributor is ex-Lancia Volumex with vacuum advance disconnected. Pipe visible between carbs is for breather. Auxiliary driveshaft is technically redundant on this engine and could have been removed (if time permitted!). Carbs are 48 DCOE on GC straight-shot manifold.



CH2/2: Engine in the car. Inlet manifold extensions had to be offset to clear the suspension leg.

CH2/3: Tom Casey holds off fellow Irish driver Neville Stanley with the new engine at Ringwood.

Test 3 results: This engine, with a IIID inlet cam (48/67) and IIIA ex cam (74/34), now yielded at 5000rpm +17lbf ft torque and at 5500rpm +3.2 lbf ft compared with *Case History No 3*, Test 5.

In addition, if the new torque results are plotted *(see graph)* against the *estimated* torque of the earlier engine the torque gains below 5000 were more impressive. As anticipated, the torque above 6000 is weaker, though this may not be significant, since by 6000rpm the car is well away from the corners, where the race is 'won and lost'.

In order to further increase the torque, it was felt that there were a number of options. Certainly when compared with *Case History No 3*, the engine with 48 carbs seemed to be at a torque disadvantage compared with 45s, except at the top end. This is partly due to the higher air velocity through 45s. With the mild exhaust cam, the short 4-1 manifold (the only one available at time of testing) was possibly causing serious loss of pressure wave effect and possible interference between cylinders at all speeds. The compression ratio and valve size were felt to be about right, but clearly there was a possible improvement to be gained from varying the inlet tract length because, all along, the engine had been extremely sensitive to cam changes and ignition timing, therefore pressure wave effects were obviously a major factor, and tuning of the inlet tract might lead to torque benefits.

Test 4 results: Substantial increase in torque over Test 3 up to about 5500rpm, giving the potential for greater acceleration out of corners. A trade-off in torque and power over approx 5500 compared with the earlier engine (*Case History No 3*, Test 5), but with a properly laid-out exhaust manifold the torque may well be enhanced at higher speeds. Clearly, the torque effect of the longer inlet manifold reaps huge dividends at low speed (compare with Test 3) – at 4000rpm +13.2lbf ft, 4500rpm +2.8lbf ft, 5000rpm +12.6lbf ft and 5500rpm +2.3lbf ft – whereas there are small, if noticeable, losses at higher speed due to the higher inlet wave frequency not working with the long inlet tract.

It is worth noting that all the tests on this engine were carried out using standard 2¼" Weber rampipes.

The serious drop in torque at 5000rpm has now been turned into the most effective result so far, presumably due to the presence of a strong reflected wave in the inlet tract.

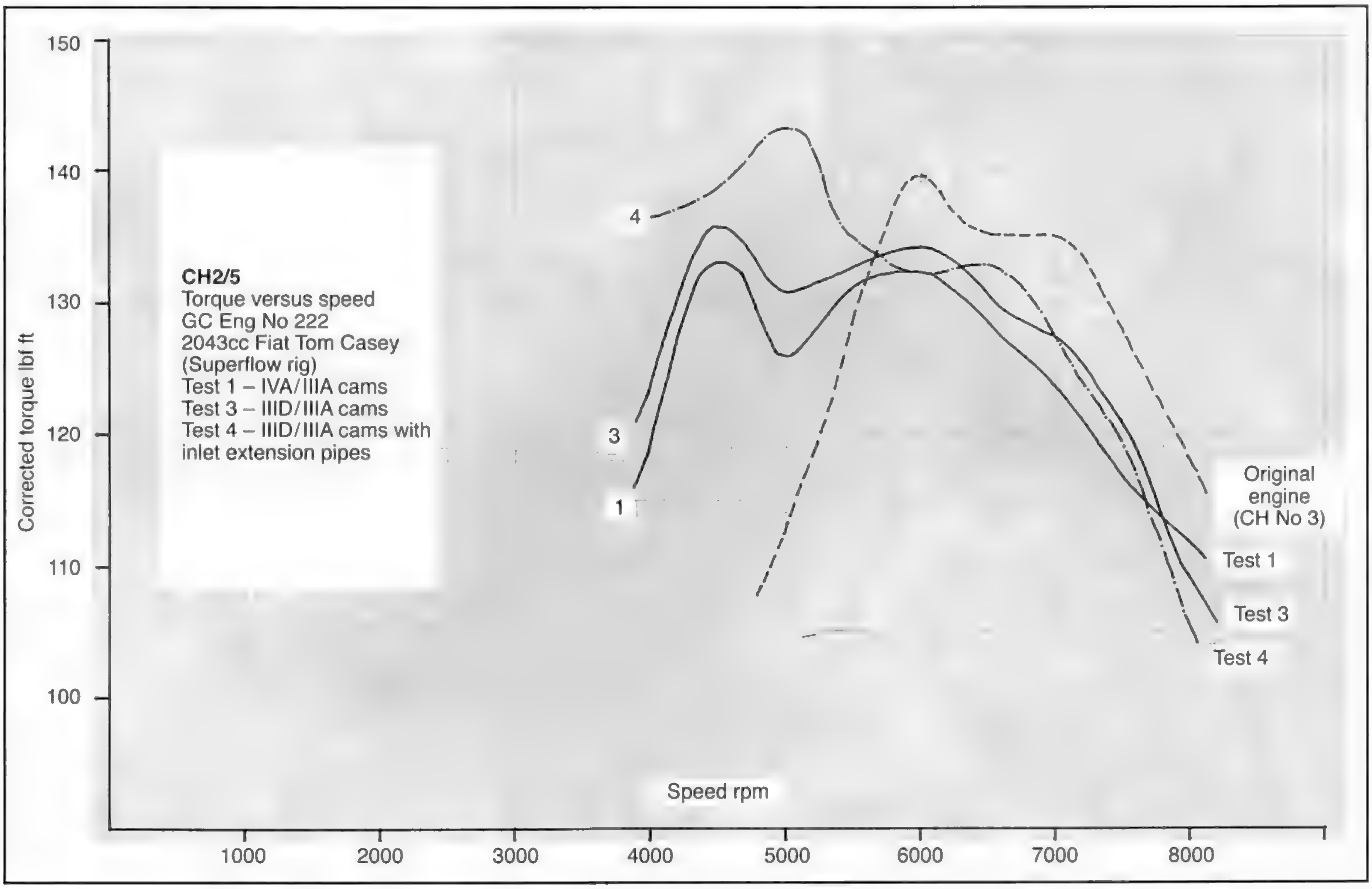
Postscript: On its debut race at the 1995 NHRA European Championship this engine took 2nd place in the 40-lap race,

TEST 4 IIID inlet cam (100°), IIIA exhaust cam (110°), 6" extension pipe fitted between manifold and carb.

SPEED (rpm)	TORQUE (lbf ft)	POWER (bhp)	BSFC (gm/kW hr)	
4000	146.5	111.6	360	JETTING:
4500	148.7	110	338	165 main
5000	153.3	146	314	155 air corrector
5500	145	151.9	298	peak torque 154lbf ft
6000	141.9	162.2	332	@ 4750rpm
6500	142.9	162.2	336	
7000	137.2	182.9	330	
7500	129.5	185	328	
8000	115.2	175.5	365	

CH2/4: 4-1 manifold used for testing Tom Casey's engine.Take-off below collector is for Lambda-Sond sensor which measures A/F ratio. This poorly made example suffered numerous stress cracks and had to be constantly repaired. Unusual shape of tailpipe highlights problem of finding manifolds which will actually fit on rig!

demolishing major opposition, including the '94 World Champion Ricky Hunn (Mass Peugeot, SHP Spaceframe). Tom Casey was only beaten by a Vauxhall 16v car (Jeff Simpson), but had he not lost six places during the second qualifying heat (due to an oil mist from a leaking oil filter cap), Tom would have probably started on pole or 2nd place in the Championship and the final result might have been a win. (In the second heat, before this mishap, Tom was bumper-to-bumper with Simpson and another 16v car and all three were drawing away from the pack.) This result was the first *major* vindication of the Fiat engine in NHRA racing.



CH2/5
Torque versus speed
GC Eng No 222
2043cc Fiat Tom Casey
(Superflow rig)
Test 1 – IVA/IIIA cams
Test 3 – IIID/IIIA cams
Test 4 – IIID/IIIA cams with inlet extension pipes

CHAPTER 6

CAMSHAFTS AND VALVE TRAIN

Cam designation and main features

The camshafts of an engine are probably the most significant contributors to development of more power (assuming a suitable CR) and have the major influence on determining the shape of the engine's torque curve.

A mass of material published in books and motorsport magazines in recent years now means that owners tend to be more aware of some of the factors affecting choice of camshafts, yet because of the *apparent* simplicity of their operation, their *true* mode of operation, with all its complex criteria, is sometimes misunderstood.

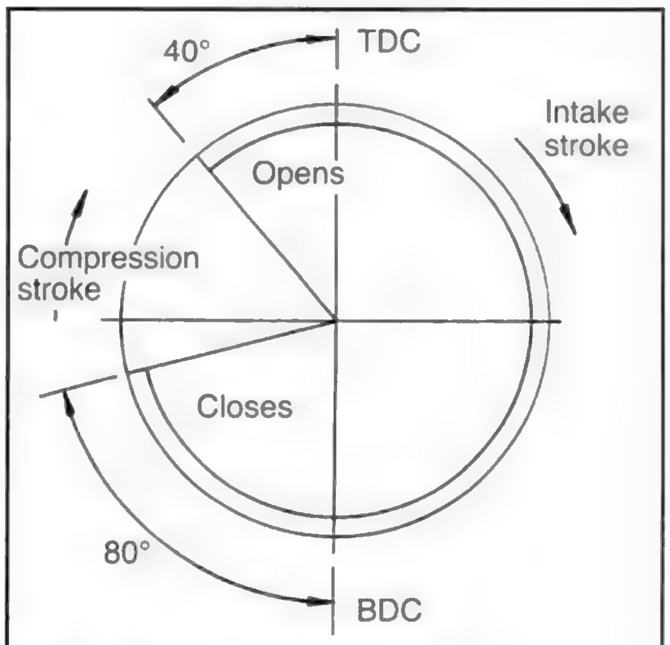
By a now accepted convention, camshafts are denoted by their timing and lift characteristics, *eg* 40/80 in, 80/40 ex, 10.8mm lift. These figures mean that the inlet cam opens 40 crank degrees before TDC and closes 80 degrees after BDC, the exhaust cam opens 80 degrees before BDC and closes 40 degrees after TDC. With competition cams, these timing figures are usually quoted with the true, or actual, running clearance between cam and tappet (shim/bucket assembly) and the lift stated is normally the 'nominal' lift, the true lift being nominal minus clearance; in other words, in the aforementioned case, if the running clearance was 0.4mm, the actual lift would be 10.4mm.

It is important to point out at this stage that Fiat conventionally define their camshaft timing with quite a wide clearance – usually 0.8mm (32thou"), whereas competition cam manufacturers' quoted cam timing is given with the true running clearance. This means that the apparently short durations quoted for Fiat/Lancia TC cams are actually longer in practice. For example:

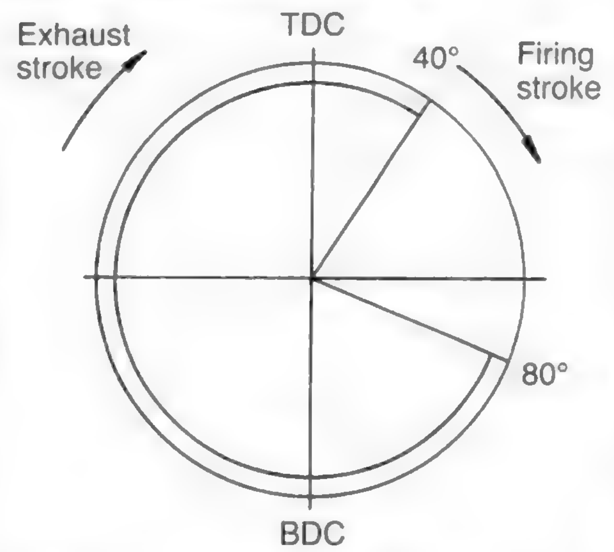
ENGINE	QUOTED FIAT/LANCIA TIMING/DURATION WITH 0.8mm CLEARANCE	TRUE DURATION WITH 18THOU" RUNNING CLEARANCE
1608	26/66 66/26 – 272° IN/EX	278°
2/BETA	13/45 (INLET) – 238°	250°
VOLUMEX	13/39 (INLET) – 232°	250°
STRADA 105TC	10/48 (INLET) – 238°	276°

Note: The figures for true duration (measured at GCT) may be slightly approximate since they were measured from 'used' cams. Fiat's designation of cam lift (*eg* 131 2*l*, 9.9mm) is the actual lift of the cam; the true lift in the engine will be 9.9mm minus the clearance **(6/1)**.

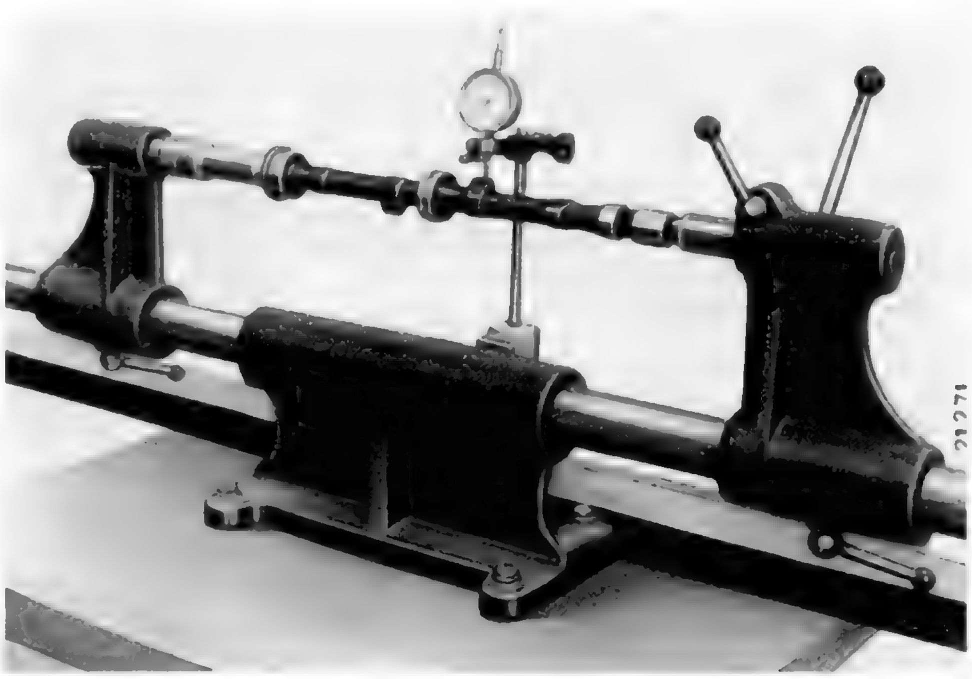
It is usual to display the cam motion as shown in diagrams **6/2** and **6/3**.



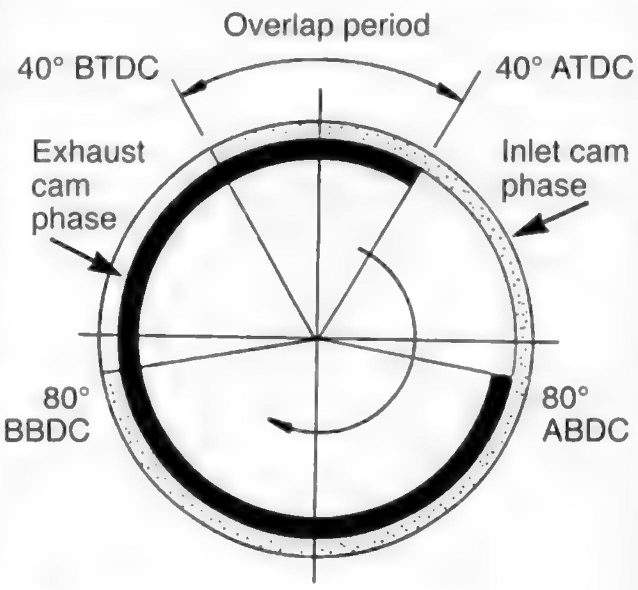
6/2: Inlet phase for 40/80 cam



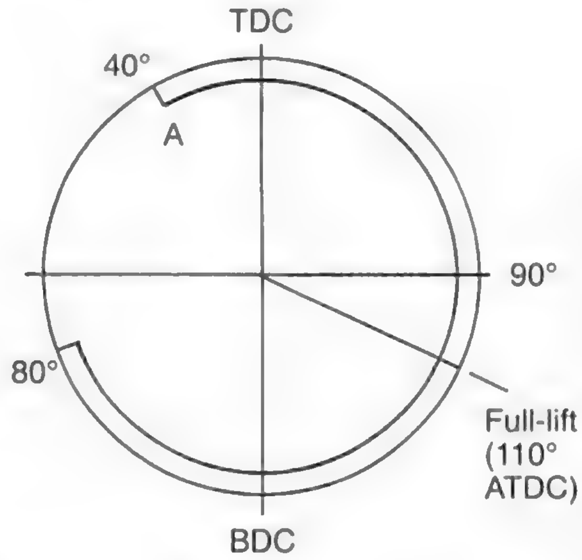
6/3: Exhaust phase for 80/40 cam



6/1: Checking lobe lift of a camshaft from a 132 1800 using magnetic base dial indicator. Fiat state both inlet and exhaust lift should be .3824in (9.714mm) and camshaft journal runout no more than .0008in (.02mm). (Fiat Auto SpA – copyright reserved)



6/4: Cam phases superimposed

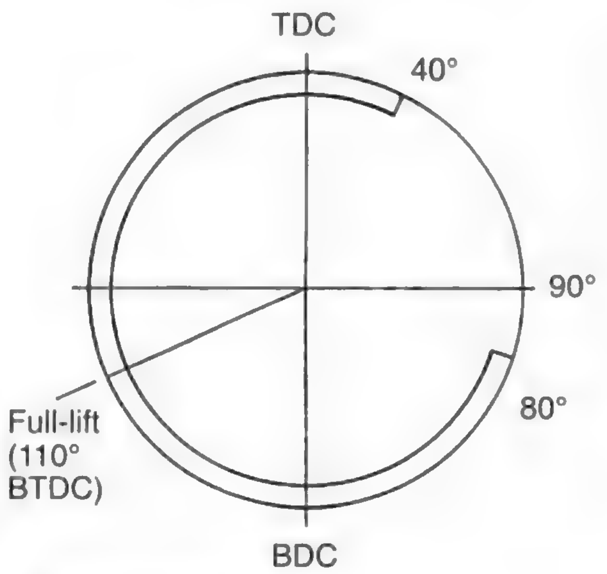


6/5: Inlet full-lift

1) A simple calculation is first used to establish geometric halfway point between 40° before TDC and 80° after BDC, ie:

$$\frac{\text{duration}}{2} = \frac{40°+80°+180°}{2} = 150°$$

2) Position of inlet full-lift relative to TDC is now calculated (full-lift is 150° from point A)
150°–40° = 110°
Hence inlet full-lift is 110° after TDC



6/6: Exhaust full-lift

1)

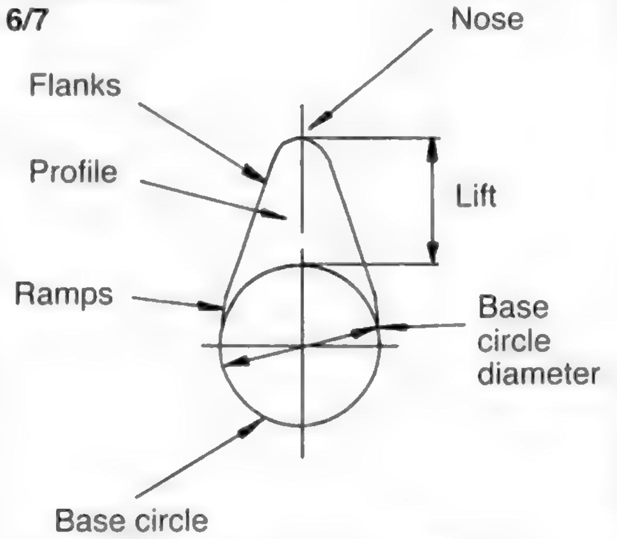
$$\frac{40°+80°+180°}{2} = 150°$$

150°–40° = 150° before TDC

As the exhaust stroke comes to an end, the inlet valve starts to open (in this case 40° before TDC) and consequently there is a period of overlap where both valves are open (its extent varies from engine to engine), and this may be illustrated by overlaying the two diagrams (see **6/4**).

The position of full lift (FL) of each camshaft can be readily determined for all standard and aftermarket cams since no 'asymmetric' profiles are currently available. It is significant since it is used to 'time up' the cams and for cam timing checks. In the 40/80, 80/40 cam example, the FL position is determined as follows (see **6/5, 6/6**):

Camshaft lift is measured by placing a dti on the base circle of the cam and rotating it until the profile is at its highest point (cam nose) (**6/7**).

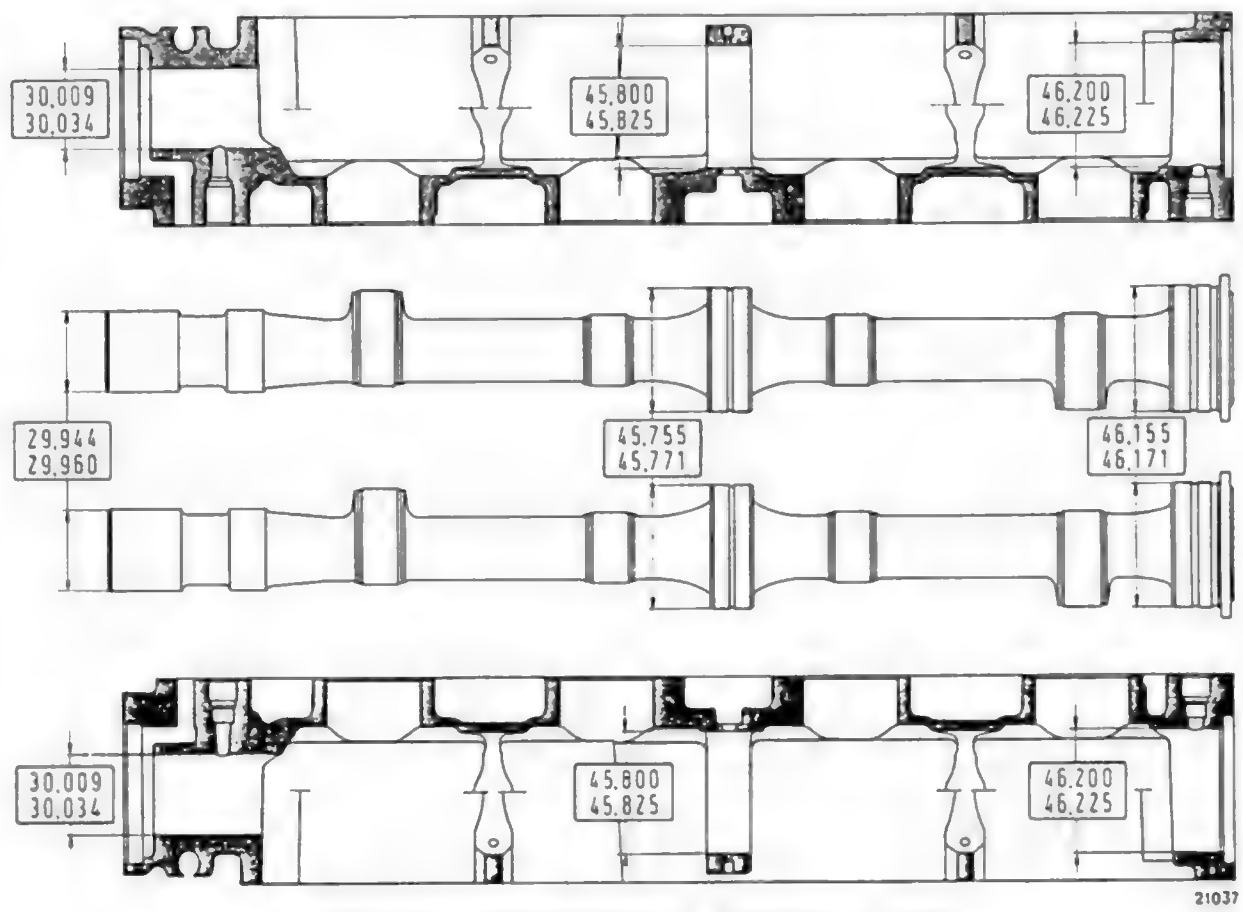


Cam designers incorporate opening and closing ramps into the profile design to minimize the accelerative and decelerative load on the train as the valve opens and closes. A harsh opening phase can cause spring and valve tip damage, and a harsh closing phase can cause the valve to bounce on its seat – leading to loss of power and impact damage. The main acceleration of the valve takes place when contact is made with the cam flank.

Camshaft duration can be measured simply using a dti, and a protractor bolted to the camshaft, measuring the number of degrees of rotation from the first point of opening (on the ramp) to final closure. The cam may be supported in a special fixture as shown (manufacturers use computerized devices for best accuracy) or supported on V-blocks. Mounting in a lathe is also useful. Cam timing is always quoted in 'crankshaft degrees'; since the cams rotate at half crank speed (this is common to all four-stroke engines since the full firing cycle for each cylinder takes 720°) the duration figure (in degrees) for the cam measured in this way must be doubled. When measuring in this way the running clearance must be allowed for to obtain the true, effective duration.

Checking standard cams

The early 8v TC is designed such that the cams and buckets run in 'oil bath' lubrication, *ie* partially immersed in oil, so



Main data of camshafts and seats in upper cylinder heads.

30.009 mm = 1.1814 in	45.825 mm = 1.8042 in	29.944 mm = 1.1788 in	45.771 mm = 1.8020 in
30.034 mm = 1.1824 in	46.200 mm = 1.8189 in	29.960 mm = 1.1795 in	46.155 mm = 1.8171 in
45.800 mm = 1.8031 in	46.225 mm = 1.8198 in	45.755 mm = 1.8013 in	46.171 mm = 1.8178 in

6/8: This extract from a Fiat 132 manual shows cam dimensional data common to all 8v TC models, early and late. (Fiat Auto SpA – copyright reserved)

that, provided the engine oil level is maintained and regular (6000-mile) oil changes are observed, wear of the valve train is quite unusual. Normal wear tends to be confined to the cam nose (some lift may be lost on a high-mileage model) together with mild wear of the three cam journals. The later, reversed-port heads dispensed with the oil bath by adding oil drain slots to the base of the cam box, presumably because it was deemed unnecessary.

Housings (there is no separate cam bearing as such) can exhibit signs of scoring and the centre and rear housings of the cam boxes (on 8V models) go slightly oval over an extended period of time. Mild bearing wear can be tolerated, but once the running clearance between the centre/rear journals and their housings exceeds about 3thou" there may be a significant loss of oil pressure. The cams on the 16v are provided with oil spray bar lubrication. The base circle of the cams should not exhibit any signs of polishing or rubbing, indicating that valve clearances are too tight. If such damage is evident there is a strong chance that the cam nose will also be damaged. **(6/8, 6/9)**

If there is any doubt as to the straightness of the cams (run-out on the centre bearing journal should not exceed $^{8}/_{10}$thou" or 0.02mm), use a dti to measure the run-out. The run-out on the cam wheel itself should not exceed 2thou" on the outer periphery of the wheel.

GCT have found over the years that the longevity of the standard cams is excellent (if only the same could be said of some aftermarket cast iron cams!) and good used cams can happily be swapped between models.

CAM LOCATION

With the exception of turbocharged and supercharged models (which have different inlet and exhaust lift) the cam design and dowel location on a given engine is identical for both cams. The variation of inlet/exhaust timing is allowed for in the design of the cam pulleys, which have timing holes (or marks) in differing places according to model. Remember that some models have distributor drives – slot or gear – on one of the cams. **(6/10)**

6/10: Distributor drives: Left to right: end-drive Beta ie, 124 Sport (exhaust-driven), Monte Carlo (inlet-driven). Cam at left is GC St IV race type. Note difference in nose profile. If Monte Carlo cam is installed in 124, distributor will run backwards – gears are not interchangeable. End-drive slots vary – make sure you have right one for your engine.

6/9: Damage to camshaft caused by use of an overly thin shim – lobe was striking bucket. Minimum shim thickness is 3.20mm. Luckily damage was spotted before serious problems developed. Note normal wear characteristic where base circle runs into ramp/flank.

Interchangeability – standard cams

Allowing for any distributor drive requirement (end drive or gears), standard cams can be swapped between models, but if the new cam lift is in excess of 0.5mm extra, dry-build the engine.

Cam swaps undertaken at GCT include:

a 131 2*l* (9.9mm lift) cams in Beta 1600 engine
b 130 TC cams (10.03mm lift) in Lancia Delta Turbo 1600 *ie*/carb (dry-build required)
c Lancia Beta 2*l* *ie* cams (9.9mm lift) in Lancia Volumex (dry-build required)
d 1608 cams (long duration, 9.5mm lift) in 131 2*l*

GCT cannot quantify the exact back-to-back power increase from these conversions as only the final power outputs (with fully modified engines) were obtained, but estimate that the increases would have been in the order of:

a 6bhp
b 25bhp
c 18bhp
d 8bhp (the increase resulting from the longer duration, although peak lift was approximately 0.4mm less)

Aspects of competition cam selection

Performance of competition cams is governed by six criteria:

1 Position of full lift
2 Overlap
3 Duration and timing (opening/closure) figures
4 Lift at TDC
5 Amount of full lift and dwell
6 Shape of nose profile (lift integral)

Position of full lift, duration and timing

The position of full lift is governed by two main factors – the accelerative rate which the valve springs can withstand, and the gas velocities through the inlet and exhaust tracts. A short-duration cam (with fewer degrees between start of lift and full lift) requires, for a given lift, a high acceleration to reach full lift earlier; this may place an unacceptable load on the springs and possibly lead to aggravated cam, valve tip and shim wear. It is more usual to base the decision of where to position full lift by examining the effect of the full lift position relative to optimum cylinder vacuum and hence torque production.

GCT have evaluated that the optimum positions for full lift vary between 100°–112° ATDC for the inlet cam and between 100°–112° BTDC for the exhaust cam. Full lift position, because of the foregoing accelerative restraint, is inextricably linked both to cam timing/duration as a whole and peak lift (obviously the higher the peak lift the greater the valve train load).

It is clear that, using the 40/80 inlet cam as an example, if the cam full lift position is altered, say to 108° ATDC from 110° ATDC, the cam will open and close 2 crank degrees earlier, thus changing the cam timing from 40/80 to 42/78. The duration of the cam will remain unaltered at 42°+78°+180° = 300°. The effect of swinging the cam timing a few degrees either way is examined extensively in *Case History No 3* (**6/11**). (It is worth noting at this stage that to do this quickly and effectively, vernier cam wheels are required.) One of the consequences that follows naturally from this revised full lift position is that the overlap is increased by 2°.

This tends to lead to more top-end power *(see Chapter 15)*. At the same time, 2° earlier closure of the inlet cam will reduce the reverse flow of fuel/air mixture back through the inlet tract at low speeds – improving torque – though at higher engine speeds the filling of the cylinder may be impaired because the inlet tract momentum carries the fuel/air into the cylinder at high speed well beyond BDC.

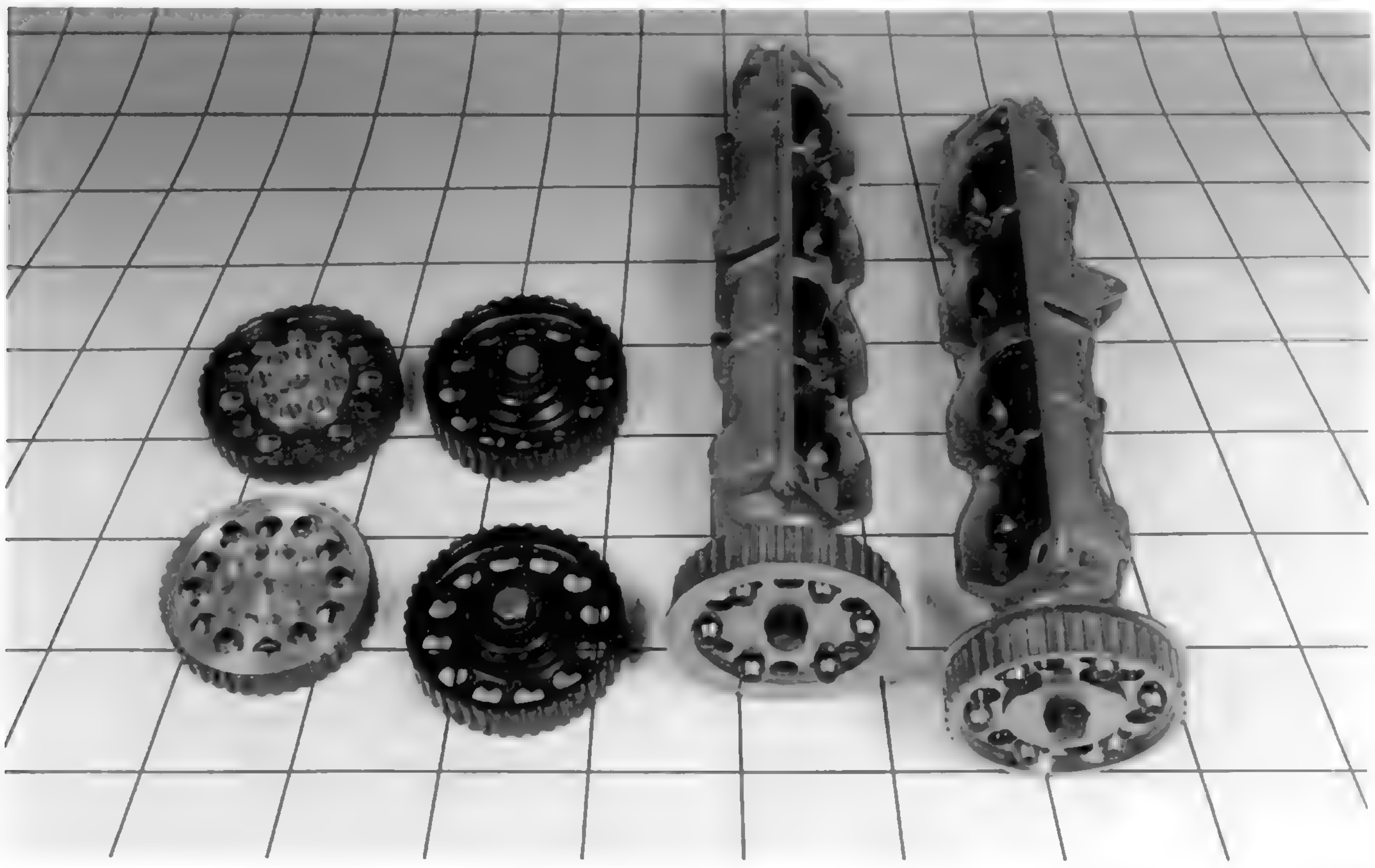
Conversely, altering the position of full lift from 110° to 112° will reduce the overlap from 80° to 78° and tend to improve bottom-end torque (when exhaust gas speed is low), but because the inlet valve is held open 2° longer (closure at 82° ABDC) the reverse-pumping effect at low speeds will mitigate against this torque benefit. However, because at very high speed the cylinder filling may be enhanced, it may readily be seen that alteration of the full lift position (as with other cam criteria) can be confusing to say the least!

[*Author's note:* Some engine builders refer to 'advancing' or 'retarding' the cam timing. This can add to the confusion, since two cams are present; at TDC on the overlap, the inlet cam is opening up, whereas the exhaust is closing down. GCT prefer to identify in the form of degrees alterations relative to full lift.]

INLET CAM TIMING AND VALVE SIZE

The magnitude of the pressure drop generated in the cylinder in the early part of the induction stroke depends partly on the residual pressure at the end of the exhaust stroke (below atmospheric pressure with a well-designed exhaust system), the valve size and cam timing. It is true to say that a small inlet valve with its low rate of opening ('curtain area' presented *versus* time – *see head section*) can generate a higher depression in the cylinder than a large one for a *given camshaft timing* – thus perhaps leading to higher port velocity and a stronger bottom-end torque characteristic. (However, the requirement to fill the cylinder quickly increases at high rpm and the larger valve is more suited to this need.)

Camshaft maximum lift is best arranged so that peak lift coincides with the point of maximum depression (several lbf/in^2 below atmospheric), usually around 100–112deg ATDC to optimize the cylinder filling at this point. Because of the high rate of opening of the large inlet valve it is worthwhile considering a delayed opening point compared with a small valve to enhance the cylinder depression as this will result in superior cylinder pressure (above atmospheric) around, and in fact well after, BDC. At high rpm, a long-duration inlet cam/big valve combination gives good torque *(see Case History – Tom Casey – 48/80 inlet cam, 46/40 valve)*, but the bottom end and mid-range can suffer unless the inlet tract is optimized (to suit the cam design) to generate high momentum at these lower speeds.

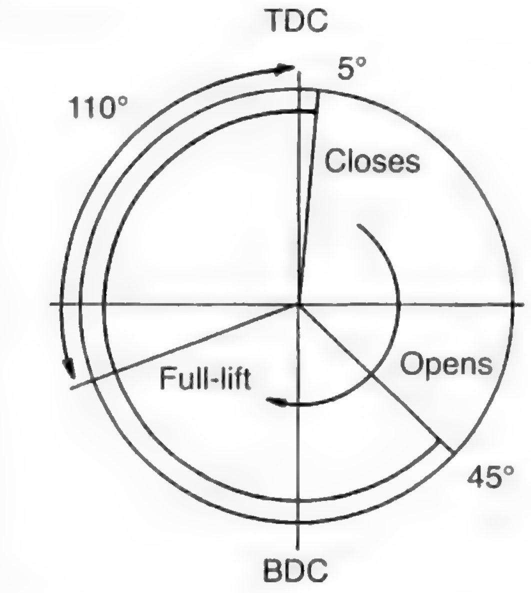


6/11: Adjustable/vernier cam pulleys. Kent Cams (left and right) and GC type (centre) flanges can be (left) machined off Kent types to accept 1" belt, but pulleys must still be fitted as shown with rear-flanged pulley on inlet side. If flanges are retained only standard ¾" belt may be used and auxiliary driveshaft pulley must have front flange. With 1" belt (GC pulleys or modified Kent types) auxiliary driveshaft pulley must have rear flange. Note that if auxiliary driveshaft pulleys are swapped, fuel pump lobe (on engines with stroke of 79.2mm-plus) may interfere with No 2 con-rod unless timing hole is repositioned or driveshaft is modified. Pulleys shown will not fit late engines (eg Delta Turbo 1600 ie, Integrale) which have smaller offset. (GC alloy types are now also available.)

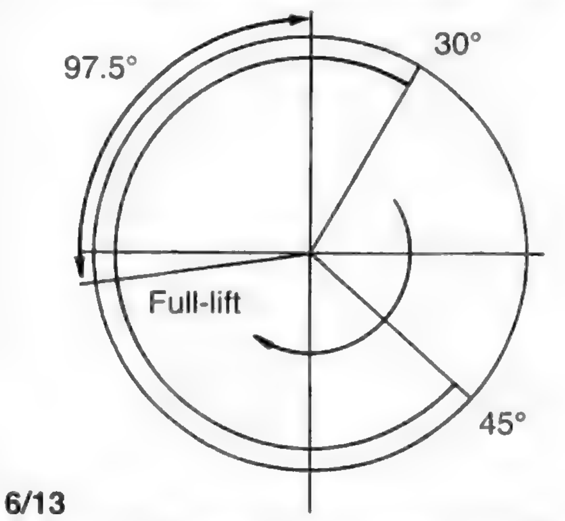
Changing the position of full lift for the exhaust cam in the above example leads to a similar set of effects. If full lift is changed from 110° to 108° BTDC, the valve is held open 2° longer (increasing the duration) and potentially increasing top-end power. However, there is a risk from this later opening of reducing the magnitude of the initial exhaust pulse (and reducing power from mid-range upwards significantly). Conversely, this later full lift position means that exhaust valve opening occurs at 78° rather than 80° BBDC and thus more useful work may be available from the power stroke. Bringing the full lift forward to, say, 112° may lead to a power increase by increasing the magnitude of the exhaust pulse and reducing overlap; remember at this stage, though, that especially with 4-1 exhaust systems, the strong pulse is utilized to enhance scavenge on the overlap, and reducing the overlap may negate this benefit.

In summary, GCT have found that altering full lift for a given cam type is very much a case of 'swings and roundabouts' – adding torque in one place and losing it in another. So sensitive is the TC to this kind of tuning that it is best carried out on an accurate, computerized dyno. Cam designers (including GCT) will determine the best full lift position to achieve valve full lift correspondent with optimum inlet/exhaust velocities. Deviating from this data is very much a question of 'trial and error'.

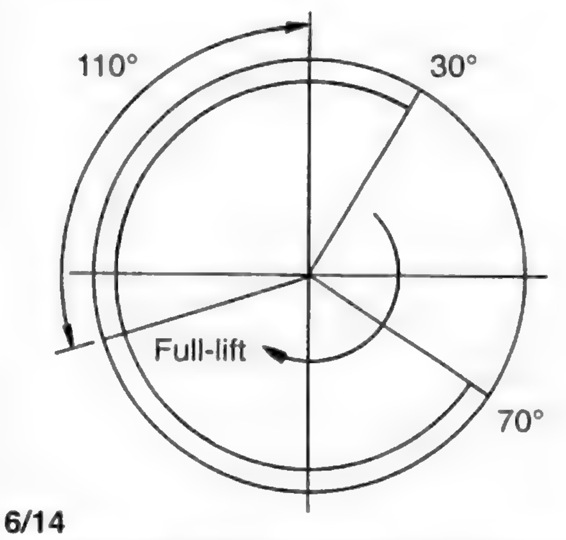
The amount of duration defines, in part, the volume of fresh charge or exhaust gas the inlet and exhaust tracts will flow (for a given lift characteristic) during the full 720° rotational cycle of the cylinder. The effectiveness of this duration for a particular engine is closely linked to the positions of valve opening and closure. Best results come from using competition cams with the highest possible accelerative rate; for medium lift (10–11mm actual) the duration can be kept relatively short on a high-acceleration profile (around 300°), but as the lift is increased the duration may have to be extended. Keeping the duration short can be used as a means of reducing the overlap period and enhancing mid-range torque – and driveability. It is worth mentioning that the recommended valve clearances can often be tightened up to extend the duration (and overlap) of the cams *(see Case History No 10)*.



6/12
A: Typical standard turbo exhaust cycle



6/13
B: Turbo exhaust valve closure extended by 25°



6/14
C: Long-duration turbo exhaust cam cycle

Turbo cam timing

In order to extract the best possible performance from turbocharged competition TCs, it is necessary to extend the exhaust cam duration to raise the turbine speed. Inevitably, with symmetric profiles, if the exhaust valve closure is delayed (allowing more high-speed exhaust gas into the turbine) full lift will have to be similarly moved unless the exhaust valve opening is advanced as well; see above (**6/12**).

In **6/13** full lift has been moved to 97.5° BTDC – which could be too late to capitalize on the strong exhaust pressure wave. A better solution would be to similarly change the 45° BBDC opening by 25°, *ie* to 70°, which, as **6/14** shows, brings full lift back to 110°.

There may be a loss of bottom-end torque because the power stroke is reduced by 25°, but there will be a strong gain at higher speeds due to the higher turbine speed at high rpm.

It is difficult at this stage to give specific rules as to the best timing/duration figures since much depends on the other factors. A summary of suitable combinations will be dealt with later.

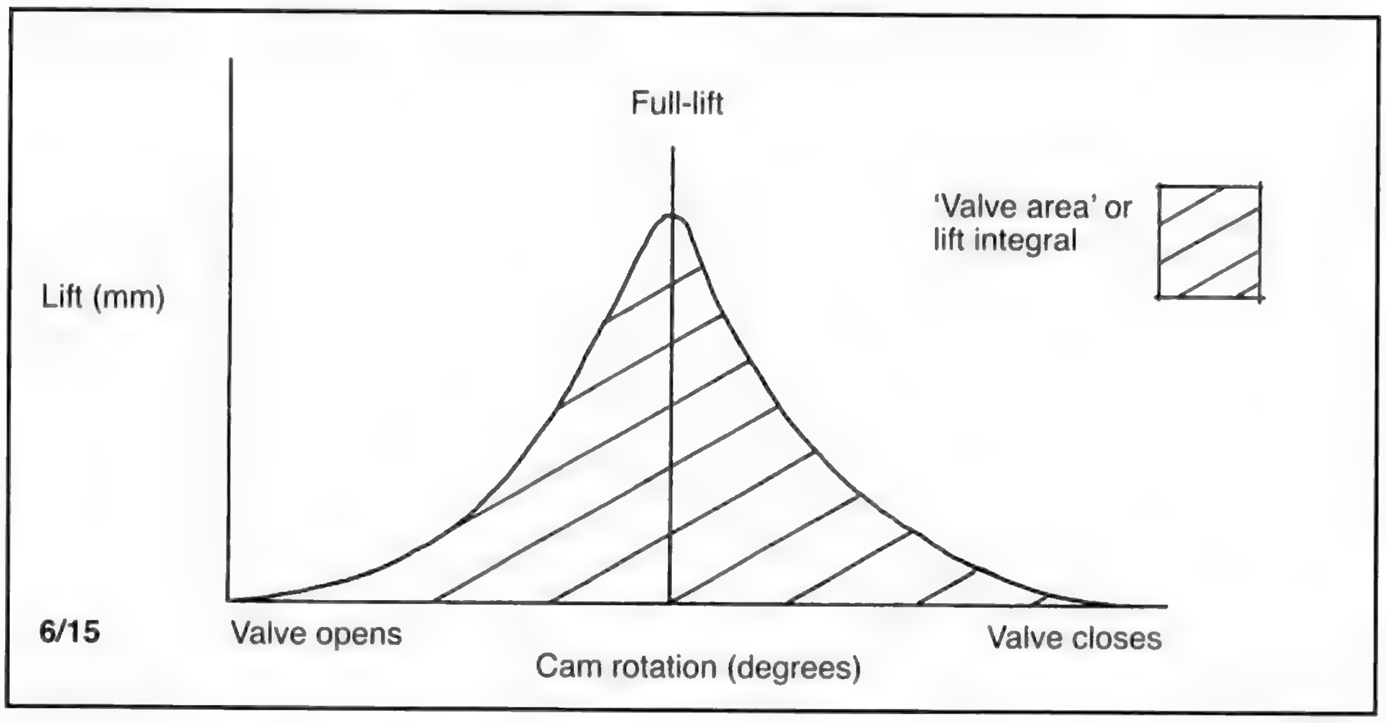
Overlap

Overlap is used for two reasons. Holding both valves open around TDC allows the depression created in the cylinder by the outgoing exhaust pressure wave and the effect of the incoming mixture to purge the cylinder of residual contamination left over from the combustion process (which would tend to weaken the incoming mixture and reduce power). On boosted engines, because the fresh charge enters under pressure, this latter criterion is less critical, but extensive experiments with quite radical inlet cams on supercharged engines (conducted in the immediate postwar period) have proved conclusively that cylinder purging can lead to quite significant power increases (albeit at the expense of fuel economy). This 'blowing-down' has the added benefit of cooling the exhaust valves.

To be effective, overlap is best used on high-speed engines. Use of excessive overlap can lead to a massive 'hole' in the torque curve on certain engines, where at low speed the exhaust gas pressure wave and gas velocity are too low and flow reversal (through the inlet port) takes place. Overlap is very much tied in with LATDC. The extent of reverse-flow is primarily governed by the inlet cam lift at TDC, and so it will be readily appreciated that on a given engine, two cams with the same extent of overlap but different LATDC, will exhibit markedly different torque characteristics.

Compression ratio is another vitally important aspect where overlap is concerned. A high CR (*eg* 10.0:1 or greater) gives rise to a very high exhaust gas pressure wave, and so will work well with long-overlap cams.

The amount of overlap used on standard and competition TCs varies greatly. Relatively low-revving (6000rpm) 2*l* turbocharged engines may have virtually no overlap (and LATDC), which gives them a very strong low-down torque response (although power drops off quite dramatically at high speed). GCT St II cams usually employ around 80° of overlap with not more than 3mm LATDC; St III rally cams may have 70°–90° with more lift, *eg* 3.5–4mm. St III race and St IV cams may have



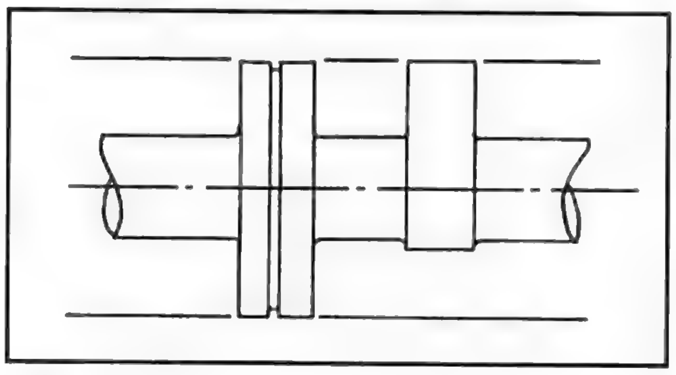
6/15

94°–100°, with over 5mm LATDC. The most radical inlet cams tried so far by GCT, for NHRA oval racing, have had overlap in the region of 104°. Standard n/a TCs, for example the 124 Abarth (1800cc), which demonstrated a quite exceptionally strong torque curve from 2500–6000rpm, have only a modest degree of overlap with only 1½–2mm LATDC.

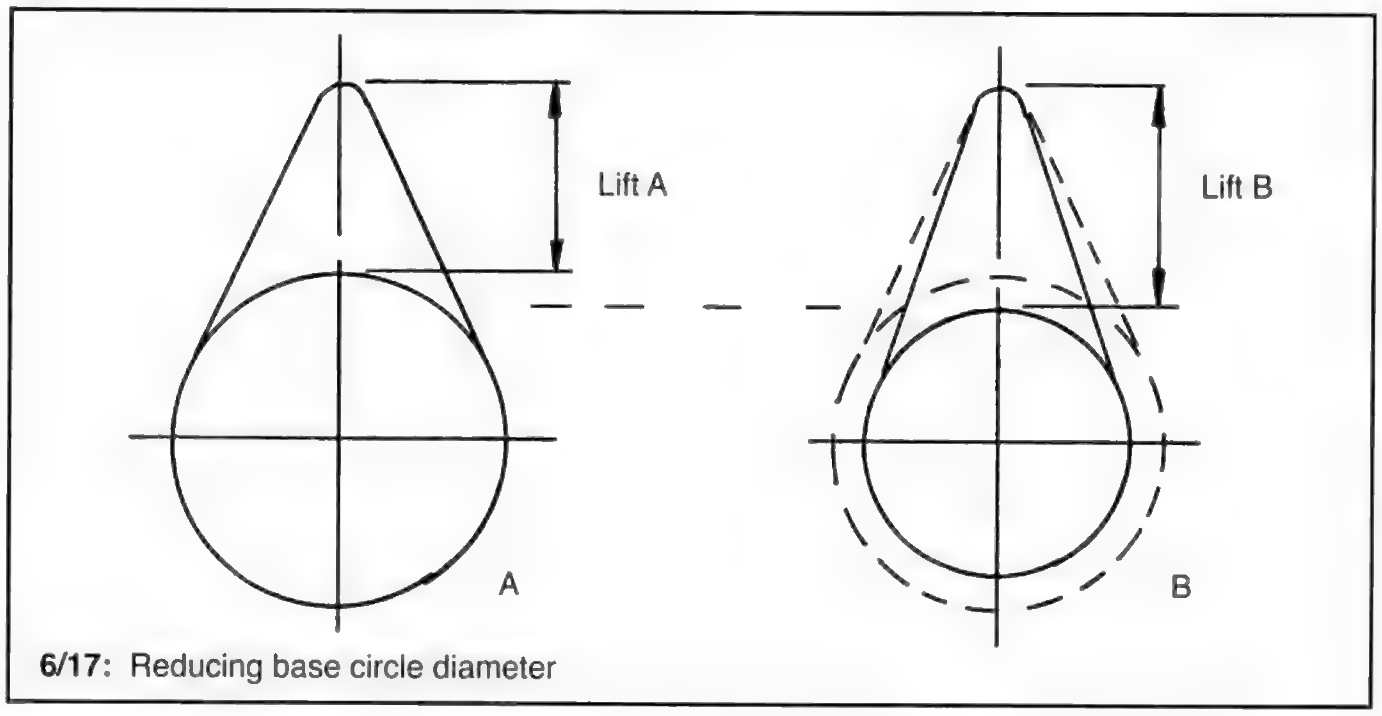
The pursuit of ultimate torque must take into account a number of factors; large overlap may not lead *per se* to this. It is largely a question of identifying where in the rev-band the torque is most needed. A high-compression TC with twin sidedraughts and quite mild cams may well give a better torque characteristic mid-range (around 4000rpm) than a comparable engine with long-overlap cams (even with low LATDC) though the torque response of the latter will probably be superior at high speed.

Turbo overlap

Turbocharged engines (on boost) respond in the same way as normally aspirated engines. Whereas a production TC may have as little as 10° overlap, a Gp A rally Integrale may require as much as 50–60°. If the exhaust cam duration is altered, as described earlier, to improve top-end turbine speed, the inlet cam duration may be similarly extended – allowing both a better high-speed torque characteristic



6/16: Showing equal radius of centre journal and profile on 8v TC.



6/17: Reducing base circle diameter

from the overlap effect and increased filling after BDC (which the turbo can now supply).

Lift and nose profile (6/15)

The area under the curve defines, to some extent, the flow-rate characteristic of the cam. Bear in mind, however, that certainly in the case of normally aspirated TCs, whether the flow-rate will match this curve depends on numerous factors such as the valve/port flowrate characteristic, exhaust performance (especially back pressure) and engine speed. In adverse conditions the flowrate may be nowhere near the optimum, which is one reason (the others being ring pressure loss and inertia/frictional losses) why torque is not constant from any engine throughout the rpm range.

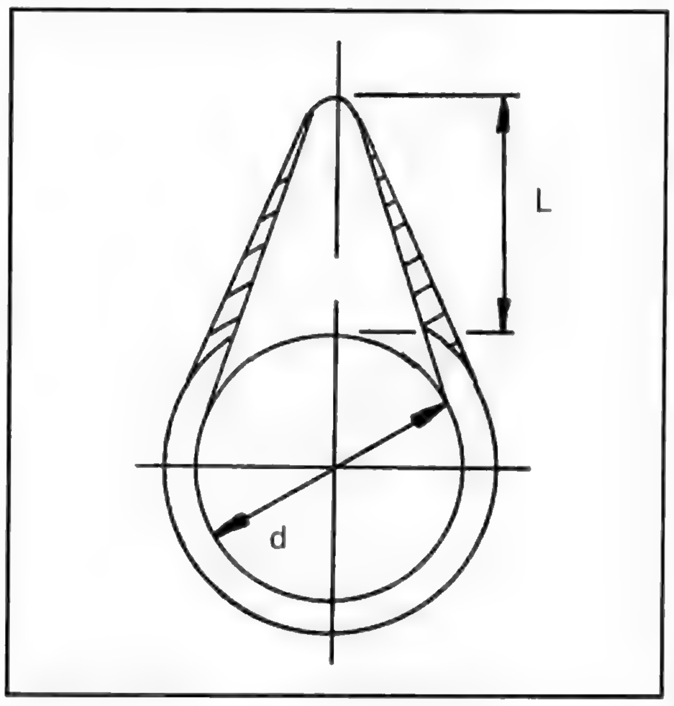
Examination of a standard 8v TC cam will show that the radius of the cam centre bearing journal is the same as the profile nose. (**6/16**)

The only way lift can be increased is by reducing the base circle diameter (**6/17**) or by increasing the height of the cam nose and bearing as well. If the nose height is increased without an oversize journal, the camshaft will not fit through its housing.

This is merely illustrative, not to say that reprofiling standard 8v cams is worthwhile. GCT experience of reprofiled 8v cams is that although the lift (and indeed duration) can be increased, the lift integral may be significantly reduced (**6/18**) and the ramps become excessively harsh.

Use of a profile ground from a billet (cast iron or steel) allows the cam designer greater flexibility since the accelerative rate of the profile can be maintained, even with a small base circle (**6/19, 6/20**).

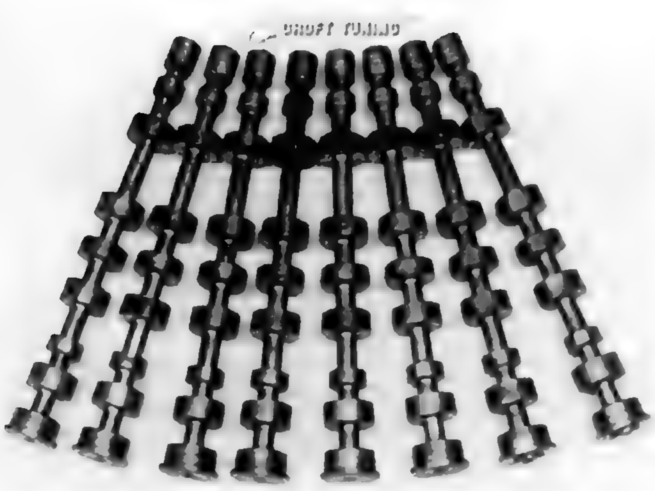
The maximum lift GCT have achieved with a standard-sized centre bearing is



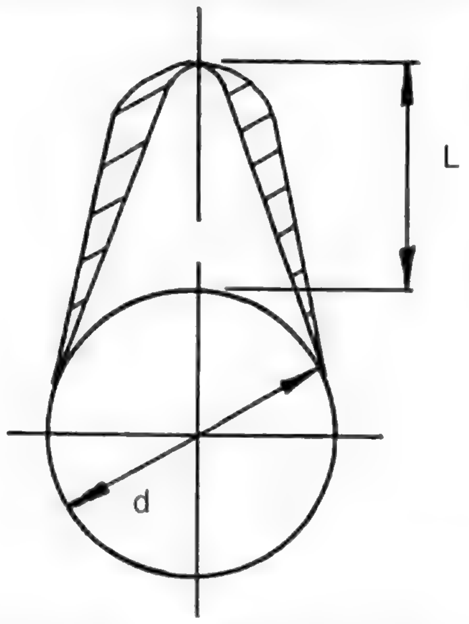
6/18: Narrow nose profile on reprofiled cam can reduce lift integral.

around 11.8mm actual. Oversize bearings will allow 12 or 13mm with a consequent increase in peak flowrate *(see graphs – head preparation – Chapter 5)*.

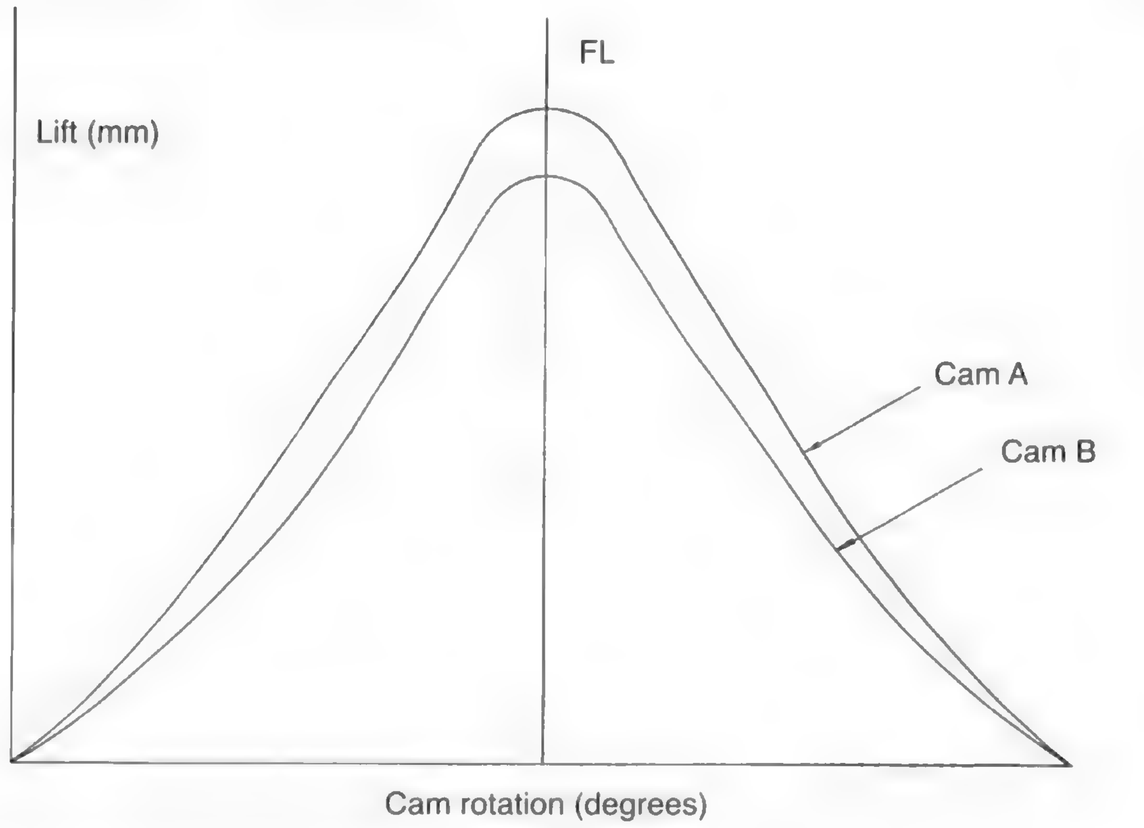
The search for high acceleration and high lift on a cam with a standard centre bearing leads naturally to a reduction in base circle diameter, and this in turn increases the LATDC. This is an



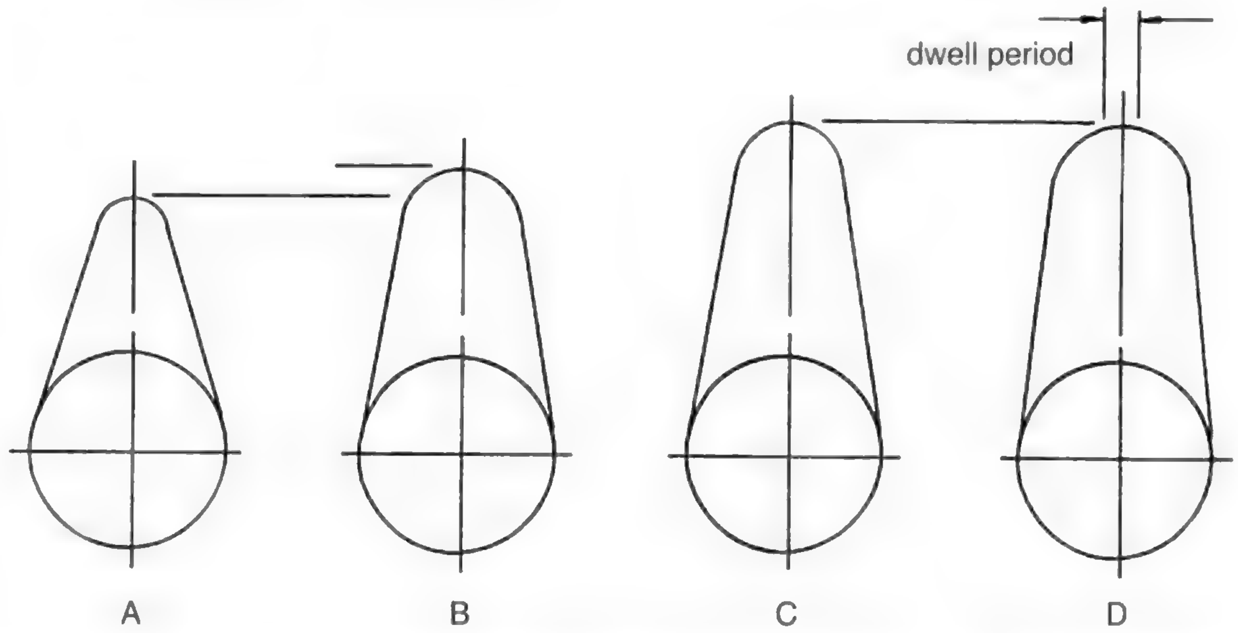
6/19: Raw steel billets machined from En 40B. Next phase is profile grinding, followed by hardening and bearing grind.



6/20: Increase of lift integral with billet cam compared with regrind of same lift and base circle diameter (L/d) (shaded area).



6/21: Lift integral for two cams of same duration but different lift.



6/22: A: Small dwell, low acceleration, medium lift
B: Long dwell, high acceleration, more lift than A
C: Short dwell, same acceleration as B, high lift
D: Increased dwell over C for same lift increases acceleration
Profiles exaggerated for clarity.

important factor, as discussed in *Case History No 2.* A more suitable torque response can often be obtained keeping the base circle fairly large (and reducing cam acceleration/lift) rather than going for a 'wild' grind that is hopelessly untractable at low revs.

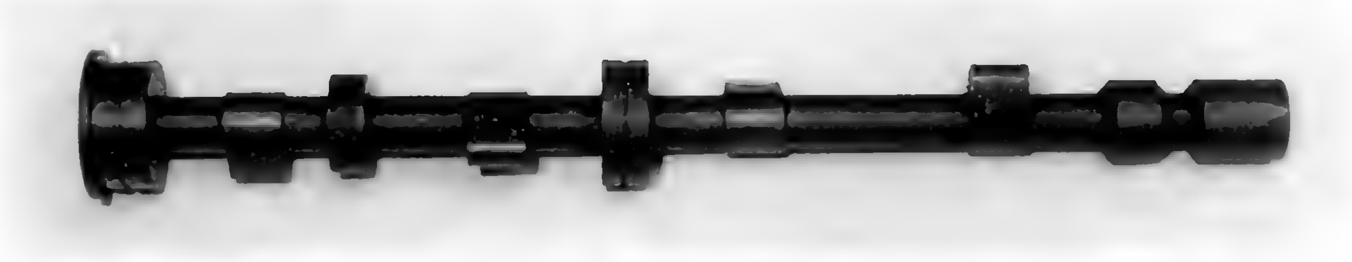
Peak lift has a significant effect on maximum torque, indeed it may be the only option left on an engine where everything else has been optimized; it is quite reasonable, having established the optimum LATDC and cam timing/duration for a particular engine, to redesign the cam purely to do this. **(6/21)**

The primary criterion affecting use of cam A will be whether the valve spring will withstand the acceleration and tolerate the lift required. The lift around TDC will determine the size of the valve pockets. GC 4A cams, for example, timed at 106° inlet full lift (ATDC), produce 4.6mm actual lift at TDC. Timed at 98°, the lift is 1mm greater, *ie* 5.6mm. (This emphasizes the crucial importance of knowing exactly how deep the valve pockets in the pistons are if a dyno-cam swap session is envisaged.) The LATDC is also significant (from a mechanical point of view) in that a radical amount of lift (*eg* +5mm LATDC with valves of 45/39mm and over) and high duration/acceleration can cause the valves to 'clash' during the overlap period; this must be allowed for during preparation of the cylinder head – the valves may need to be recessed deeper into the head – possibly leading to an increased intruder section in the pistons to keep the CR high enough.

Because the charge/exhaust velocity will be high around full lift, it is common for competition cams to have a significant dwell period around full lift (**6/22**). This may vary from a few degrees on mild (St II) cams to as much as 6° on full-race versions. Mechanically, there is no problem with this as the piston is still a long way from the inlet and exhaust valves, except that a long dwell phase puts a very high loading on the cam nose because of the problem of blending the required shape into the cam flank. The design of the dwell phase is heavily dependent on the cam acceleration/lift characteristic and capabilities of the valve train.

Peak lift selection for the inlet cam can be made on the basis of what best matches the cfm flow characteristic of the cylinder head, but as the graphs show in *Chapter 5*, the lift can be very high indeed before flow starts to diminish! Exhaust cam selection should take into account the peak flowrate criterion discussed in

Chapter 5 in that there may be a measurable benefit from reducing exhaust peak lift (and using a 'cam-mix, *ie* different inlet/exhaust profiles), but at the same time perhaps increasing the accelerative rate.



6/23: GC lightweight EN 40B steel cam (St III rally) manufactured from billet. Process involves raw machining, lobe grinding, hardening and bearing grind. Raised section between rear two profiles allows for gear cutting for top-mounted distributor types with special hobbing cutter. En 40B is an ideal cam material because its distortion after final heat treatment (Tufftriding/nitriding) is minimal. It has outstanding durability and tensile strength – especially important for cams with large lift integral.

Camshaft performance/selection

Production TC cams are designed to give maximum flexibility (the ability to pull full-throttle from low speeds in a range of gears) combined with acceptable top-end performance (giving reasonable engine life). To this end, the lift at TDC is kept to the bare minimum whilst the peak lift is reasonably high. Overlapping is quite modest (nominally 30° maximum on the longest-duration model – 15/55 55/15 timing – if the unusual 1608 cams are neglected), but the valve layout of the TC, with the large opening area presented across the valves, seems to ensure good cylinder scavenging nonetheless. Indeed, a testament to the effectiveness of the layout is the success of the GC Fast Road 2*l* engines (of which over 60 had been built at time of writing) utilizing merely a raised CR (9.6:1), ported/blueprinted head and twin 45 carbs (36 choke), which have all produced 148–158bhp with standard cams.

Modest LATDC is achieved by having a large base circle, high peak lift by giving the cams a fairly large centre bearing, allowing the cam nose to be quite tall. The lift integral (valve area) and acceleration are mild by competition standards. Whatever is done to the engine (except for turbocharging) the production cams will not tend to 'peak' much higher than 6500rpm (and usually between 6000–6300rpm), but in this range will produce a good driveable torque characteristic.

However, such a low peak power/rpm figure will not satisfy the serious competitor. Unless the engine has forced induction, even the 2*l* version can never be made to produce more than about 144lbf ft of torque with *standard* cams, in any specification. A simple reckoning will lead the reader to the obvious conclusion that, since the torque is a curve rather than a horizontal line, even the best peak torque characteristic, if it is confined to low rpm, will not satisfy the requirements both of high top speed and high road speed in each gear (leading to rapid acceleration).

By their very nature, competition cams

1600 ENGINES (N/A) – COMPETITION CAM SELECTION (GC recommendations)

	USE	CR	HEAD	FUEL SYSTEM	EXH	CAMS	OUTPUT
A	ROAD	9:1	PORTED, STD VALVES	STD (CARB)	STD	STD	AROUND 110BHP, PEAK TORQUE/POWER POSITIONS APPROX 300RPM HIGHER THAN STANDARD
B	ROAD	9:1	PORTED, STD VALVES	STD (CARB)	STD	2*l*	AROUND 116BHP, PEAK TORQUE/POWER POSITIONS APPROX 300RPM HIGHER THAN STANDARD
C	ROAD	9:1	PORTED, STD VALVES	TWIN 40s (32 CHOKE)	STD	2*l*	AROUND 125BHP, PEAK POWER AND TORQUE APPROX 500RPM HIGHER
D	ROAD	9:1	PORTED, STD VALVES	TWIN 40s (34 CHOKE)	STD	290°–300° DURATION, LESS THAN 3.5mm LAT DC, NOT MORE THAN 10.5mm ACTUAL LIFT MILD LIFT INTEGRAL	AROUND 135BHP @ 7000RPM, MAX TORQUE AROUND 5500–6000RPM. POSSIBLE WEAK TORQUE BELOW 4000 DEPENDING ON CAM DUE TO LOW CR
E	ST II ROAD/ RACE OR RALLY	9.6–10:1	PORTED, STD VALVES	40s (34 CHOKE)	4-2-1	AS D ABOVE BUT MEDIUM LIFT INTEGRAL	AROUND 145BHP @ 7500RPM, MAX TORQUE AROUND 5500–6000 (BETTER BOTTOM-END TORQUE THAN D)
F	ST III RALLY	10.5:1 (FORGED PISTONS)	44/36 VALVES PORTED	45s (36 CHOKE)	4-2-1	AS D BUT WITH MAXIMUM LIFT INTEGRAL AND UP TO 310° DURATION	AROUND 160BHP @ 8000RPM, MAX TORQUE AROUND 6500RPM, WEAK TORQUE BELOW 4500RPM
G	ST IV RACE	11:1 (FORGED PISTONS)	44/36 VALVES PORTED	45s (36 CHOKE)	4-1	4–4.5 LATDC 300–310° DURATION 10.9 ACTUAL LIFT AND MAXIMUM LIFT INTEGRAL	AROUND 170–175BHP @ 9000RPM, MAX TORQUE 7000RPM. WEAK TORQUE BELOW 5000RPM

NOTES: These recommendations apply to 1592, 1608, 1585 including Delta/105 TC big-valve 43½ inlet types. There will be approximately 5% variation on outputs/rpm depending on bore/stroke/valve size. For Weber-Marelli injection add 10% to bhp (approx).

2*l* ENGINES (N/A)							
	USE	CR	HEAD	FUEL SYSTEM	EXH	CAMS	OUTPUT
A	ROAD	9:1	PORTED, STD VALVES	STD	STD	STD	AROUND 120BHP @ 6000RPM, MAX TORQUE AROUND 3600
B	ROAD	9:1	PORTED, STD VALVES	STD	STD	AS 1600 (D)	AROUND 130BHP @ 6500RPM, MAX TORQUE AROUND 4000RPM, POSSIBLE JETTING PROBLEMS IF CAMS TOO 'WILD', STRONG, FLAT TORQUE CHARACTERISTIC UP TO APPROX 6700RPM
C	ROAD	9:1	PORTED, STD VALVES	TWIN 40s (34 CHOKE)	STD	AS B ABOVE	AROUND 140BHP @ 6800RPM, MAX TORQUE AT APPROX 5000RPM, VERY STRONG BOTTOM-END TORQUE
D	ROAD	9.6:1	PORTED, STD VALVES	TWIN 45s (34 CHOKE)	STD	STD	AROUND 150BHP @ 6300RPM, MAX TORQUE 3800RPM, VERY FLAT, STRONG TORQUE RESPONSE UP TO 6600RPM, VERY GOOD PART-THROTTLE ECONOMY, WILL PULL FULL THROTTLE (ANY GEAR) FROM AROUND 1800RPM
E	ST II	9.6:1	PORTED, STD VALVES	TWIN 45s (38 CHOKE)	4-2-1	AS 1600(E)	AROUND 160–165BHP @ 6800–7000RPM, MAX TORQUE AROUND 5000–5500RPM. LESS TORQUE BELOW 4000RPM THAN D
F	ST II	10:1	PORTED, STD VALVES	TWIN 45s (38 CHOKE)	4-2-1	AS 1600(F)	AROUND 170–174BHP @ 7000–7200RPM, MAX TORQUE AS E, WILL PULL FULL-THROTTLE FROM AROUND 2800RPM
G	ST II	10:1	PORTED, 44/36 VALVES	TWIN 45s (38 CHOKE)	4-2-1	AS 1600(F)	APPROX 178–182BHP @ 7000–7200RPM, MAX TORQUE APPROX 5500RPM
H	ST III RALLY (FOREST)	10.5–11:1 (FORGED PISTONS)	PORTED 44/36 (OR 38) VALVES	45s (40 CHOKE)	4-2-1	UP TO 4mm LATDC 209–295° DURATION, 10.5mm ACTUAL LIFT, MAX LIFT INTEGRAL	APPROX 184BHP @ 7000–7200RPM, MAX TORQUE AROUND 5500RPM
J	ST III RALLY (TARMAC) AND OVAL	10.5–11:1 (FORGED PISTONS)	46/38 (OR 40) VALVES, PORTED	45s (40 CHOKE)	4-1	UP TO 4.5mm LATDC 300°–320° DURATION, 10.8–11.8mm ACTUAL LIFT, MAX LIFT INTEGRAL	AT LEAST 190BHP @ 7500RPM, MAX TORQUE AROUND 5500–6500RPM DEPENDING ON CAM DURATION
K	ST IV RACE (CIRCUIT)	10.5–11:1 (FORGED PISTONS)	46/40 VALVES, PORTED	45s (40 CHOKE)	4-1	5–5.5mm LATDC, 295°–325° DURATION, 11.8–12.2mm ACTUAL LIFT, MAX LIFT INTEGRAL	APPROX 192–196BHP @ 7600RPM, MAX TORQUE AROUND 6000–6500RPM
L	ST IV RACE (CIRCUIT)	10.5–11:1 (FORGED PISTONS)	46/40 VALVES, PORTED	48s (40 CHOKE)	4-1	5–5.5mm LATDC, 295°–325° DURATION, 11.8–12.2mm ACTUAL LIFT, MAX LIFT INTEGRAL	198–210BHP @ 7600RPM, PEAK TORQUE AROUND 6500RPM, LESS TORQUE BELOW 6000RPM THAN K

NOTES: As 1600.
Remember that using, for example, 2*l* type 'J' cams will work perfectly satisfactorily with smaller valves, *eg* 44/38 or even 43½/36, but the peak power and torque will be less due to the lower airflow through the smaller valve throat.

'add on' at the top and 'take away' at the bottom of the rev-band. This phenomenon is allied closely to the performance of the inlet and exhaust tracts, with the chaotic performance of the various filling and exhausting pressure waves present. To this extent, the selection of a competition cam combination for the TC must be examined closely in relation to the intended use – maximum speed (road and engine) in each gear and acceleration will be critical to the competition success of the vehicle. The cams must produce the right torque characteristic in the right place.

Essentially, a number of simple components will define the selection process: lift at TDC, overlap (in degrees), peak lift, duration and the size of the cams profile (lift integral).

Valve springs and caps

Standard springs start to resonate around 7200rpm with competition cams because the low natural frequency of the springs cannot withstand the accelerative load induced by the profile. Reprofiled cams can similarly lead to spring damage because of the harsh ramps the regrinding process produces.

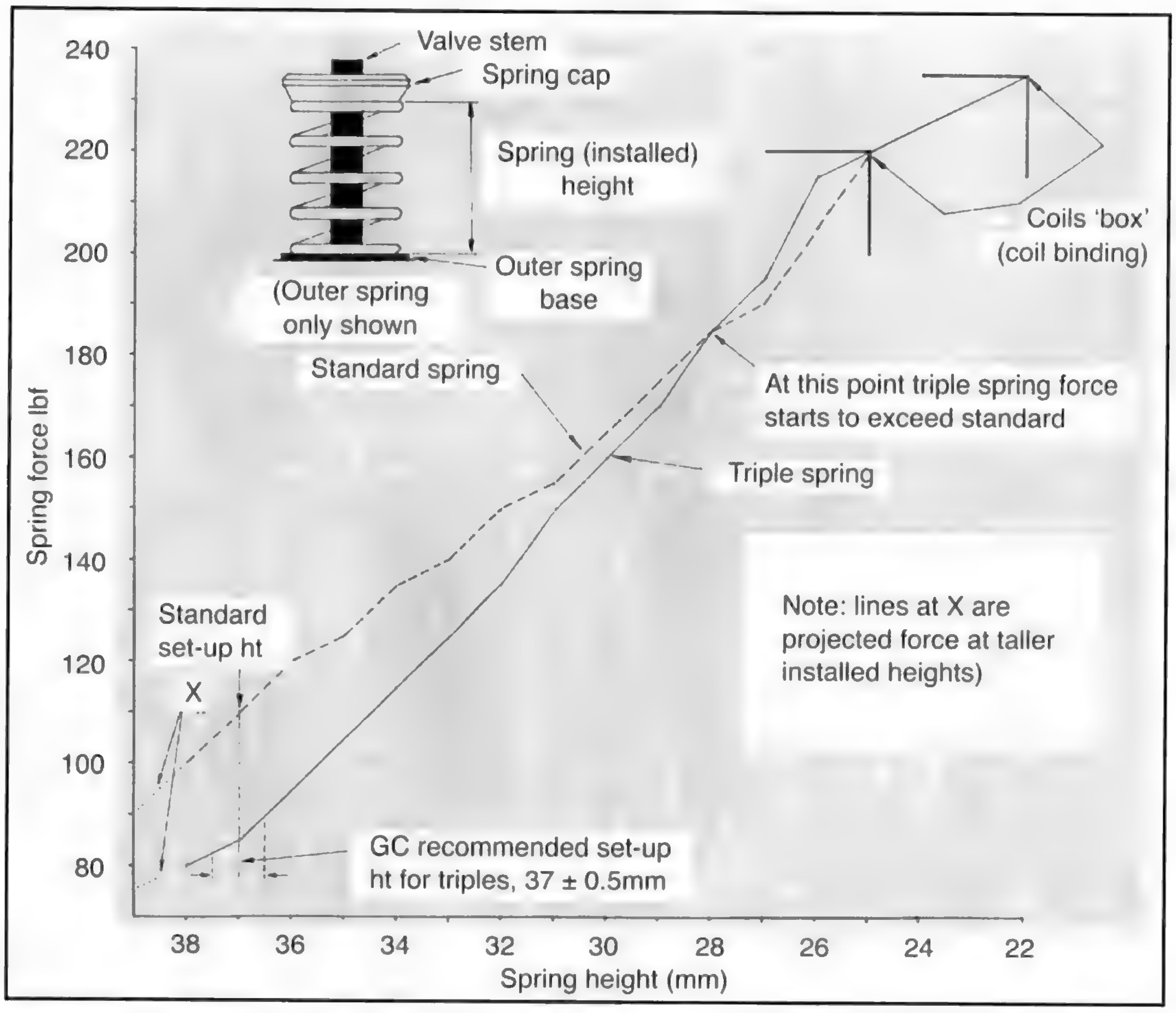
The process of resonance is quite complex, but essentially vibration is set up in the spring, causing adjacent coils to collide and the valve/bucket to come off the profile (valve float). This leads to power loss and can allow the valve to hit the piston. One way of overcoming this effect is to use harder springs with a higher natural frequency, or springs which are an 'interference' fit inside each other and exhibit a damping effect.

6/24: Front: Standard springs and caps. Safe to around 7000rpm with cams up to around 11mm lift. Rear: Triple interference springs with alloy caps. Safe to 9500rpm-plus, depending on cam, valve type. (Remember that piston strength – if cast – may be limiting factor.)

Excessively hard springs can cause the valve tip to 'peen' over. Poor spring control or an adverse closing ramp can cause the valves to bounce on their seats – leading to loss of compression and valve/seat damage.

The great advantage of the GC triple springs is that they have a relatively low nose poundage (at installed height), but their poundage (lbf/in of compression) increases as the middle (flat) spring starts to compress (*ie* as the acceleration of the cam increases). Springs are usually defined in terms of their free length, fitted length and poundage. Fiat standard springs are the same on all models (including 16v) and appropriate workshop manual data is given as follows:
Outer spring free length – 53.9mm (2.12")
Inner spring free length – 41.8mm (1.65")

The poundage of these springs can be calculated (as they have a linear load/compression relationship).
Fiat quote:
Outer spring length at load 85lbf = 36mm (1.42")
Inner spring length at load 33lbf = 31mm (1.22")
Poundage:
Outer spring compresses 0.7" under 85lbf
poundage = 121lbf/in
Inner spring compresses 0.43" under 33lbf
poundage = 77lbf/in
Hence the total poundage of the pair is 121+77 = 198lbf/in. This compares reasonably well with the test data that follows.
The poundage of GC triples varies as shown in graph **6/25**:



6/25: Graph of spring force versus compressed fit.

Notes on graph of spring force *versus* compressed height:
The spring rate of the triples is higher than that of the standard type, indicated by the steeper gradient.
The standard spring, being longer in the uncompressed state, however, exerts more force than the triple until the 28mm point is reached, where the triple overtakes it.
The nose poundage, *ie* the force exerted on the cam at the start of lift, is determined by the installed, or 'set-up' height of the spring and at 37mm is:
standard – 110lbf
triple – 85lbf
The spring rates, in terms of the force exerted relative to the amount of compression, are (approx):
standard – 206lbf/in
triple – 254lbf/in
The standard spring conforms more or less with the linear relationship expected of a constant-section, dual-wire spring; the triple has a similar characteristic until the flat damper spring behaviour upsets this. The non-linear performance resulting is a combination of the tensile strength of the damper spring and its frictional interference with the inner and outer coils.
Coil binding occurs at 25mm/220lbf for the standard spring and 22mm/235lbf for the triple.
The force measurements are taken with both inner and outer spring bases fitted on the standard spring, for the triple only the outer spring base has been used. The inner spring base can be used with the triples, but the maximum lift must be checked to ensure the coils do not box. The inner base can be left out with no detrimental effect, indeed the triple is designed to operate on a flat base. GCT have not encountered any problems of interference of the triple type with the inside wall of the cam bucket in this layout.
GCT recommend leaving 1mm between the closest coils at maximum lift with both springs; based on this, the maximum allowable lifts are:

standard spring with standard bases –
38mm installed ht/12mm
37mm installed ht/11mm
triple spring with outer base only –
38mm installed ht/15mm
37mm installed ht/14mm
This data was derived from tests using a standard-pattern spring cap. Remember to check, at full lift, that the cap does not interfere with the valve stem seal.

Production springs are installed at 37mm ± 0.5 (the actual height of the outer spring when fitted, not including the cap and spring base). For competition use (with any cam over 10.5mm lift and over 7200rpm) triple springs are strongly recommended. In the Mk1 hydroplane, the 1800 engine was revved to an incredible 10,500rpm with no valve problems, and no GC engine equipped with these springs has ever 'dropped a valve'. A gap of 1mm must be allowed between the closest pair of coils on full compression (including the inner coil). This may be simply checked by compressing the springs in a vice with the valve cap and inner spring seat fitted and measuring between the coils with a feeler gauge. 'Coil binding', which occurs when all coils touch, can lead to spring failure, and indeed may even prevent the cam from rotating. The installed height of springs may be varied safely between 36–38mm, depending on the cam lift. There is no virtue in going below 37mm, but if the installed height of the valve collet is such that 36mm cannot be achieved (because the valves are too short or the valve seat has been installed too high), the spring base area of the head may need to be relieved by machining.

Despite their odd appearance, triples retain their interference effect for two or more seasons. Springs should be checked visually on a rebuild, and their poundage checked using a valve spring tester. If the poundage or free length is more than about 5–10% below specification, even if the springs appear visually OK, discard them.

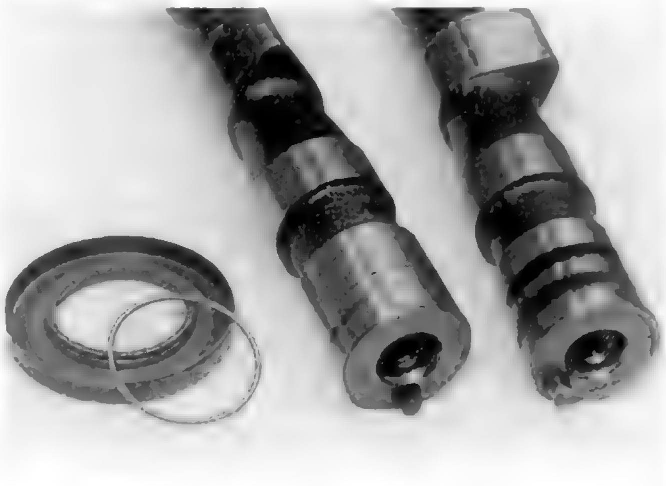
Cam buckets and caps

Although some engine builders do prefer to replace the standard bucket with lighter items (discarding the removable Fiat shim and using a lash cap – or top-hat shim – on top of the valve to obtain the necessary clearance) GCT have never encountered a problem with them either with very high-lift cams or high rpm. (Use of excessively hard springs can cause damage to the inside of the bucket, as can a valve tip which has been shortened by grinding and the contact face is not accurately at 90° to the valve stem.) A shim can only come out if chronic valve float is present (even the thinnest – 3.2mm – is located in a recess around 2mm deep – four times the valve running clearance).

The standard collets and caps are extremely durable and can be used over and again. GCT normally employ alloy (anodized HE30) valve caps over 7500rpm because they are inexpensive to make and around one-third of the weight of the standard items. They will last two seasons or more (if properly designed, with the appropriate radii in key areas), but should be checked each season. Impact wear is quite normal; presence of a crack means the cap must be discarded. (It is important to note that, if the valve is shortened *(see shimming-up section)*, the bucket must bear on the valve tip, not the cap.)

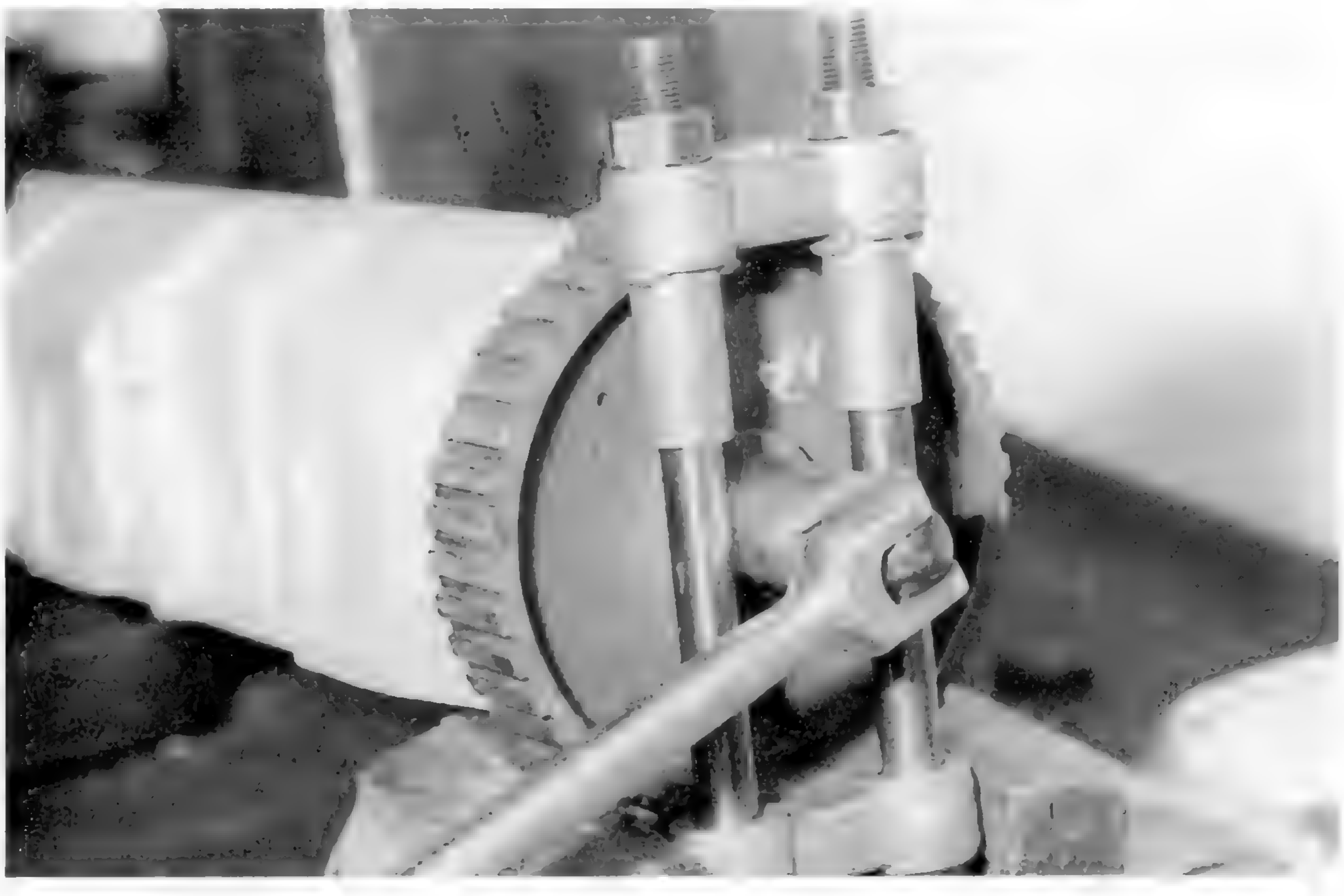
GCT have never resorted to new cam boxes/buckets (the running clearance is around 1–1½thou"), preferring to find good used ones if wear is extreme. If the bucket is too loose in its housing, it can tilt in service (due to the wiping action of the cam profile), leading to aggravated guide wear (cast iron guides may crack).

Fitting cams – standard



6/26: Fitting cams 1
If rebuilding the engine with 'used' cams, dress-up front of cam with 1000-grade emery paper to avoid seal damage. If old seal has worn a groove in cam nose, remove seal spiral spring, split ends (screwed together) and shorten by approx ½mm to exert more seal pressure.

6/27: Fitting cams 2
Best cam seals on 8v are late pattern from Tipo, which have rubber outer seal on casing. Otherwise, coat case with silicon sealant and tap gently into place. Nothing more sophisticated is needed to fit all engine seals.



6/28: Fitting cams 3
Lubricate cam bearings with oil or graphite grease, oil front seal and gently ease cam into place. If seal lip appears to catch on cam nose, use suitably rounded spike to fit into place. Ensure spiral spring does not become dislodged. Tighten camwheel bolt as shown, provided you are sure dowel holes will not have to be redrilled. If a lot has been machined off block or head, or cam boxes have been machined, or competition cams are being used without vernier or adjustable cam wheels, leave tightening till later. It is good practice to mark bolts with red paint after final tightening – no red paint means bolts are still loose. If you don't have special tool shown, steel pulleys can safely be clamped in vice – but be careful with plastic pulleys not to split flanges.

Fitting cams – non-standard
Principles of setting up camshafts (symmetric cams)

It is normally necessary to time-up the Twin-Cam with the head off. It is very easy to bend the valves if an attempt is made to move the cams independent of each other on the engine with the belt off. Do not attempt to time the cams at their full lift position (as with a single-cam engine) since it is almost impossible to assess the correct position of the exhaust cam when the inlet is at full lift so that the belt can be fitted. In attempting such a procedure it would be hard to prevent opposing valves from 'fighting' each other!

Since the Twin-Cam is a four-stroke engine, pistons 1 and 4 arrive at TDC together twice during the working cycle of the engine, once during the compression stroke (both valves shut) and once during the exhaust stroke – when the valves are on the 'overlap' (both valves open – though on some turbocharged engines there may be little or no overlap). The engine should be timed up on the overlap, *ie* the belt is fitted with pistons 1 and 4 at TDC, with the cams in their respective positions, *ie* inlet opening up and exhaust nearly closed.

To establish cam position at TDC

This is easy enough. Study the cam timing diagrams shown. The cam represented has timing figures of: 40°/80° (inlet), 80°/40° (exhaust)

The full lift position shown is simply the arithmetic halfway position between 'cam starts to open' and 'cam is shut'. Full lift position is calculated as follows:

Inlet full lift $40° + 180° + 80° = 300°$

$$\frac{300°}{2} = 150°,\ 150° - 40° = 110°$$

Hence inlet full lift = <u>110° after TDC</u>

Exhaust full lift $80° + 180° + 40° = 300°$

$$\frac{300°}{2} = 150°,\ 150° - 40° = 110°$$

Hence exhaust full lift = <u>110° before TDC</u>

The cams rotate at half crank speed, so the relative positions of the cams at TDC are:

$$\text{Inlet} = \frac{110°}{2} = \underline{55° \text{ before full lift (A)}}$$

$$\text{Exhaust} = \frac{110°}{2} = \underline{55° \text{ after full lift (B)}}$$

As this example was based on a 40/80, 80/40 cam it will be necessary to calculate figures (A) and (B) for the actual cams used.

Timing and marking cams

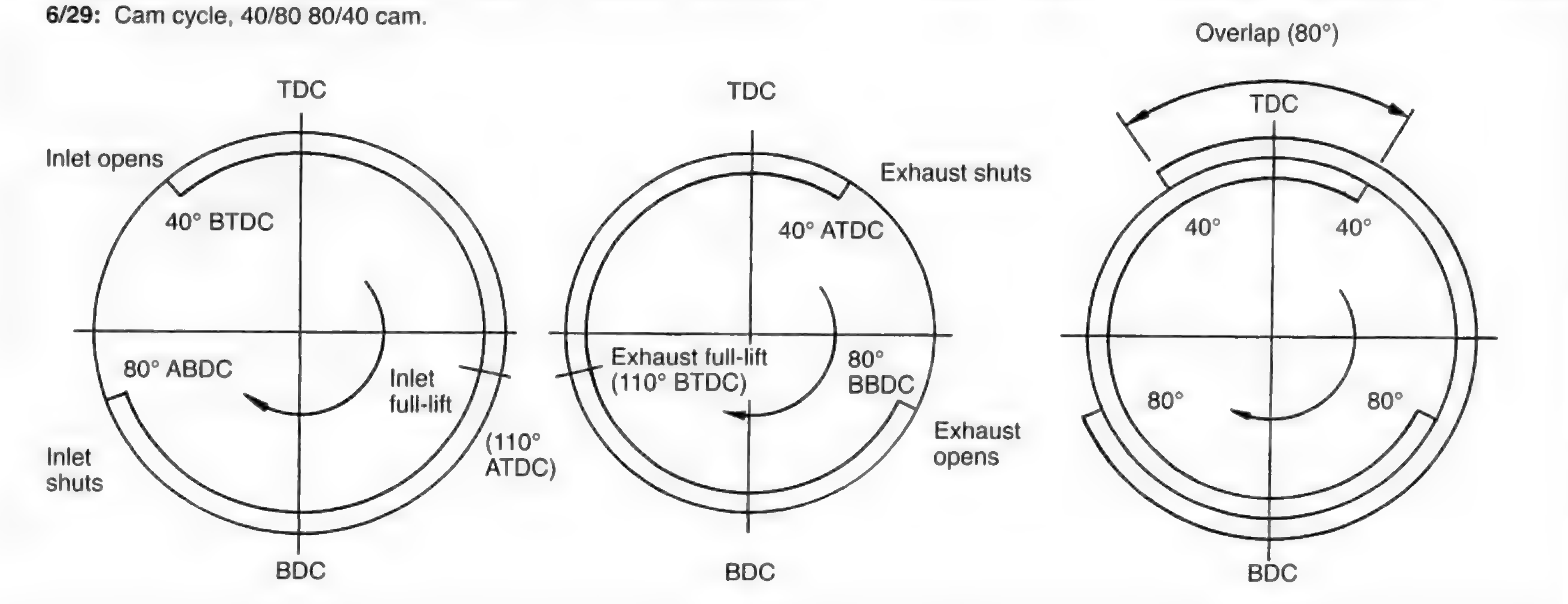
It is preferable to mark the cams with the cam boxes off the head, although this is not essential. Fit the cams to their boxes (with new seals in position) and attach the respective camwheels. It may be necessary (if verniers are not used) to redrill the camwheels or cams at a later stage when the belt is about to be fitted, so do not tighten them fully yet (but *do not forget to tighten them later!*). A permanent mark is needed on the cam itself for the same reason. A preferred method is two dots on the cam rear bearing (white for full lift and red for TDC position) and one white dot on the cam box bearing housing for lining-up.

Turn each cam box upside-down, and with a bucket in No 1 bore, establish the inlet and exhaust full lift positions with a dial gauge. Cams have a small period of dwell around full lift (**6/30, 6/31**), so do this carefully. Mark the cam and box with full lift marks (white to white).

Then attach a small (*eg* 5") circular protractor to each cam wheel with Blu Tack and fix a wire pointer to the cam box (Plasticine is ideal for this). Rotate the inlet cam anti-clockwise (thus effectively closing the inlet valve) from full lift by amount (A). Turn the exhaust cam clockwise (closing the exhaust valve) by amount (B) from full lift. Mark the cams accordingly (red to white).

These are the positions the cams must

6/29: Cam cycle, 40/80 80/40 cam.



be in when the belt is fitted (1 and 4 pistons at TDC).

Adjustable or vernier pulleys – when to use them

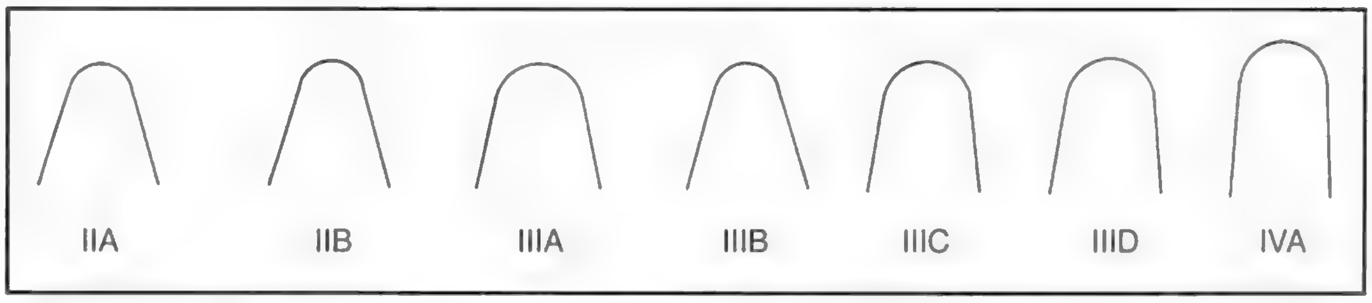
Machining more than 5thou" off the block and head (though the late-pattern head gasket – *eg* 130 TC – is 11thou" thicker than the early type and will restore the camshaft height relative to the crank),

6/30: Fitting cams 4
TRUE FULL LIFT: DIALLING-IN CAMS
Set cam at approx full-lift as shown. Turn cam clockwise so dti reading drops about 0.2mm and note dti reading…

6/31: …in this case 0.2mm. Fit protractor to camwheel, with pointer (Blu-tack will hold in place), note degree reading. Now turn camwheel anti-clockwise and through full-lift, back to same dti reading above. Note degree reading. True full-lift will be halfway between readings. Because of cam dwell around full-lift, this degree method is an accurate way of finding full-lift since lift readings are taken off cam flanks rather than nose; ±0.5 cam degrees (±1° crank) should be the tolerance to aim for.

CAM PROBLEMS	CAUSE OR EFFECT
INCORRECT TIMING	Inlet cam full lift too early – top-end power may be good but poor bottom end Inlet cam full lift too late – reverse of above plus loud intake noise/fuel standoff Exhaust cam full lift too early/late – weak torque throughout the range
INCORRECT VALVE CLEARANCES	Cam wear Irregular running Poor compression (valve not closing)
LOSS OF POWER IN PREDICTED POWER BAND	Valve float (wrong springs) Cam timing slipped (pulleys loose/belt jumped)
ENGINE WILL NOT TICKOVER BELOW 800–1000RPM	Cams too radical Cams timed-up wrongly
VERY WEAK MID-RANGE TORQUE	CR too low for cams Weak cam lift integral Too much overlap/LATDC Carburettors too big
TORQUE TOO HIGH IN REV-BAND	Excessive LATDC Excessive exhaust/inlet duration
POOR MAXIMUM TORQUE	Peak lift too low, weak lift integral
TORQUE TOO LOW IN REV-BAND	LATDC too low Lift integral too weak Duration too short Carburettors too small
VALVE-PISTON DAMAGE	Inadequate piston-valve clearances Valve float, cam timing slipped (stones?)
VALVE-SEAT DAMAGE	Mixture too rich (carbon damage) Valve bounce
VALVE-TIP DAMAGE	Springs too hard – valves not 'topped' correctly Cam profile too harsh – worn buckets
EXCESSIVE CAM WEAR	Poor quality cam material/hardening Lubrication problem Lack of start-up lubricant Coil binding Clearances too tight
EXCESSIVE CAM WHEEL RUNOUT	Cam/pulley bent Pulley not seated correctly on cam Cam incorrectly machined
EXCESSIVE CAM HOUSING WEAR	Lubrication problem Belt too tight

or alterations to the standard cam box set-up (more than one gasket, machining, or no gaskets) may alter the position of the timing marks on the cam pulleys. In this case, it is worth considering the use of non-standard pulleys. (Adjustable pulleys can be secured to the cam once it has been installed. GC verniers must not be tightened until the dowel position has been established.) When the cams are in



6/32: Nose profiles of GC cams. The bulk on the nose profile will indicate the sort of lift integral that can be expected.

6/33: Use of the wrong valve springs led to the total demise of the (low-grade) valve at left; 21-4n valve at right only bent.

their 'timed-up' (TDC) position, the belt teeth may not line up with the pulleys, so these two types allow the outer ring of the cam wheel to be rotated to suit. If it is intended to 'swing the cam timing' during dyno-testing, GC verniers have the distinct advantage that the cam timing can be accurately changed 1° at a time – a very difficult thing to do otherwise because of the problem of marking the pulley/housing with such a small graduation. GCT have not experienced problems, however, with 'bolt-up' type adjustable cam wheels (provided they are properly tightened!).

GC CAM DATA

CAM TYPE, DESIGNATION AND NOTES ARE FOR GUIDANCE ONLY, EXACT CHARACTERISTICS WILL DEPEND ON TOTAL ENGINE SPECIFICATION

No	TYPE	TIMING (true deg)	TIMED UP AT	TRUE LIFT (mm)	DURATION (true deg)	NOSE PROFILE	VALVE CLEAR-ANCE	VALVE SPRINGS
GC IIA	ST II/ Fast Road	44/84 IN 84/48 EX	IN FL 110° EX FL 110°	9.9 9.9	308°	IIA	16thou" (0.4mm)	STD or triple
Notes: Peak torque 5500rpm, peak power 7200rpm, broad spread of torque, very tractable from 2000rpm, needs CR 9:1 minimum								
GC IIB	ST II Rally Clubman	41/81 IN 81/41 EX	IN FL 110° EX FL 110°	10.2 10.2	302°	IIB	16thou" (0.4mm)	STD or triple
Notes: Characteristics similar to above but stronger mid-range torque, CR 9.5:1 minimum								
GC IIIA	ST III Forest Rally	39/81 IN 79/39 EX	IN FL 111° EX FL 110°	10.4 10.2	IN 300° EX 298°	IIIA	IN 16thou" (0.4mm) EX 18thou" (0.6mm)	STD or triple
Notes: Peak power 7000rpm, peak torque 5500rpm, pull very strongly from 2800rpm, exceptionally tractable, CR 9.6:1 minimum. More torque through range than IIB								
GC IIIB	ST III Race/ Tarmac Rally	44/84 IN 84/44 EX	IN FL 110° EX FL 110°	10.2 10.2	308°	IIIB	12thou" (0.3mm)	Triple
Notes: Peak power 7500rpm, peak torque similar to profile IIB but stronger throughout range, very tractable, CR 10:1 minimum								
GC IIIC	ST III Race/ Tarmac Rally	40/80 IN 80/40 EX	IN FL 110° EX FL 110°	10.5 10.5	300°	IIIC	16thou" (0.4mm)	Triple
Notes: Peak power 7750rpm, peak torque 6750rpm, very strong mid-range and top-end torque, reasonably tractable from 3500rpm, CR 10.5:1 minimum								
GC IIID	ST III Race/ Sprint	54/73 IN 71.5/52.5 EX	IN FL 99.5° EX FL 99.5°	10.9 10.7	IN 307° EX 304°	IIID	IN 16thou" (0.4mm) EX 18thou" (0.6mm)	Triple
Notes: Similar to profile IIIC but better mid-range torque, peak torque 5500rpm, peak power 7500rpm, not tractable below 3000rpm, CR 10.5:1 minimum								
GC IVA	ST IV Race	48/80 IN 80/48 EX	IN FL 106° EX FL 106°	11.8 11.8	308°	IVA	16thou" (0.4mm)	Triple
Notes: Peak power 7600rpm, peak torque 6500rpm, not tractable below 5000rpm!, CR 11:1 minimum								

CAM CHARACTERISTICS ARE BASED ON USE IN 2/N/A ENGINE – CAMS WILL BE 'HOTTER' IN SMALLER ENGINES.

CASE HISTORY No 3

Owner	Tom Casey
Engine No	n/a
Type	Fiat 2043cc
Use	National Hot Rod
Tested	Warrior Automotive, Aug '94
Rig	Superflow

This engine was originally built by Tom Casey using parts from GCT and raced successfully in National Hot Rod during 1994, taking Tom to 4th place in the NHRA World Championship at Ipswich. The original spec comprised:
Forged pistons 10.5:1 CR.
Race valves 46/39, triple interference springs, alloy valve caps. GC IVA cams.
Dry-sump lubrication, steel flywheel, 7¼in clutch.
Straight-shot inlet manifold, 48 DCOE, 40 choke (NHRA rule), Facet Red Top pump.
Marelli electronic ignition (end-drive distributor).
NGK B9EGV race plugs, 4-1 24" 45mm ID exhaust manifold.
The engine proved exceptionally reliable, despite running at 100°C for the entire World Championship. After the race, the engine was dyno-tested at Warrior Automotive on their Superflow rig to establish its output and examine the effect of altering the cam timing. The following results were obtained (Test 1):

In order to optimize the jetting to give around 300–330bsfc the test was re-run with 37° adv, 180 main jets to enrich the mid-range and 185 air correction to keep the top end sufficiently lean (Tests 2 and 3).

Because of the over-rich top end mixture, the air correctors were then changed to 190 (Test 4):

Flexibility test: engine pulled full-throttle down to 3500rpm.

Flexibility test: Engine pulled full-throttle from 3350rpm (Test 5).

Conclusions: It had been expected that the retarded inlet cam on Tests 3 and 4 would lead to a more significant torque increase by reducing the lift at TDC, but the modest gain in torque above 7500 was cancelled out by torque losses lower down, probably due to the pumping loss on the compression stroke because the valve was held open 4 crank degrees longer.

By opening the exhaust valve 4 crank degrees later (Test 5) and thereby effectively extending the power stroke, torque was increased *versus* Test 4 at 7000 and below, but as expected, the reduced lift at TDC reduced top-end torque. Because of the high cam lift around TDC, and the danger of piston-valve contact, and insensitivity of the engine to cam changes, it was decided to conclude tests.

TEST 1 Cams timed at 102° inlet, 110° exhaust

SPEED (rpm)	TORQUE (lbf ft)	POWER (bhp)	BSFC (gm/kW hr)	
5000	125.3	119.3	343	170 main jet, oil temp 84°C
5500	140.0	146.7	292	F16 emulsion tube
6000	150.4 max	171.9	274	water temp 69°C
6500	145.9	180.6	278	180 air corrector
7000	144.7	192.9	288	45 pump jet
7500	137.1	195.9 max	321	40° adv @ 5500
8000	invalid result			

TEST 2 Cams at 102° inlet, 110° exhaust

SPEED (rpm)	TORQUE (lbf ft)	POWER (bhp)	BSFC (gm/kW hr)	
5000	119.1	113.4	379	RESULTS:
5500	137.6	144.2	326	Peak torque increased by 1.4lbf ft
6000	151.8 MAX	173.5	286	Max power increased by 1.4bhp
6500	146.1	180.9	296	Loss of 6.2lbf ft @ 5500
7000	145.1	193.5	305	37° adv
7500	138.1	197.3 MAX	317	180° main jet
8000	127.1	193.7	314	185 air corrector

TEST 3 Cams at 106° inlet, 110° exhaust (*ie* inlet cam retarded 4 cam degrees)

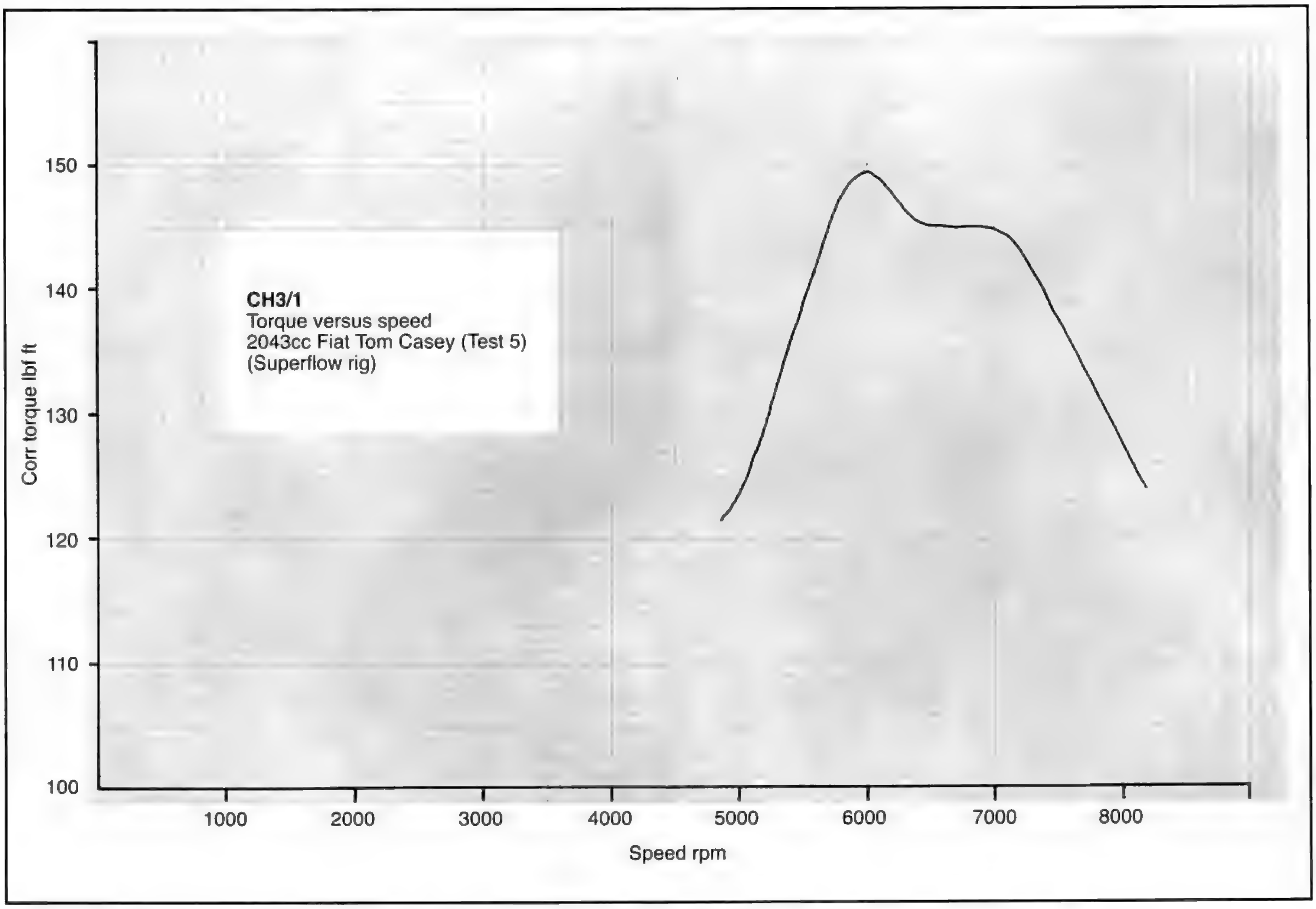
SPEED (rpm)	TORQUE (lbf ft)	POWER (bhp)	BSFC (gm/kW hr)	
5000	119.1	113.4	382	COMPARED TO TEST 2
5500	135.5	142	309	RESULTS:
6000	149.3 max	170.6	283	Peak torque decreased by 2.5lbf ft
6500	146.5	181.4	286	Peak power decreased by 1.6bhp
7000	144.7	192.9	298	37° adv
7500	137	195.7 max	326	180° main jet
8000	127.2	193.8	345	185 air corrector

TEST 4 Cams at 106° inlet, 110° exhaust

SPEED (rpm)	TORQUE (lbf ft)	POWER (bhp)	BSFC (gm/kW hr)	
5500	130.5	136.7	336	37° adv
6000	149	170.3	300	180 main jet
6500	144.6	179	285	190 air corrector
7000	144.7 max	192.9	293	COMPARED WITH TEST 2
7500	138.7	198.1	315	RESULTS: Marginal
8000	129.6	197.5	336	increase of 2.5lbf ft @ 8000
8500	93.2	150	473	Loss of torque below 7000

TEST 5 Cams at 106° inlet, 106° exhaust

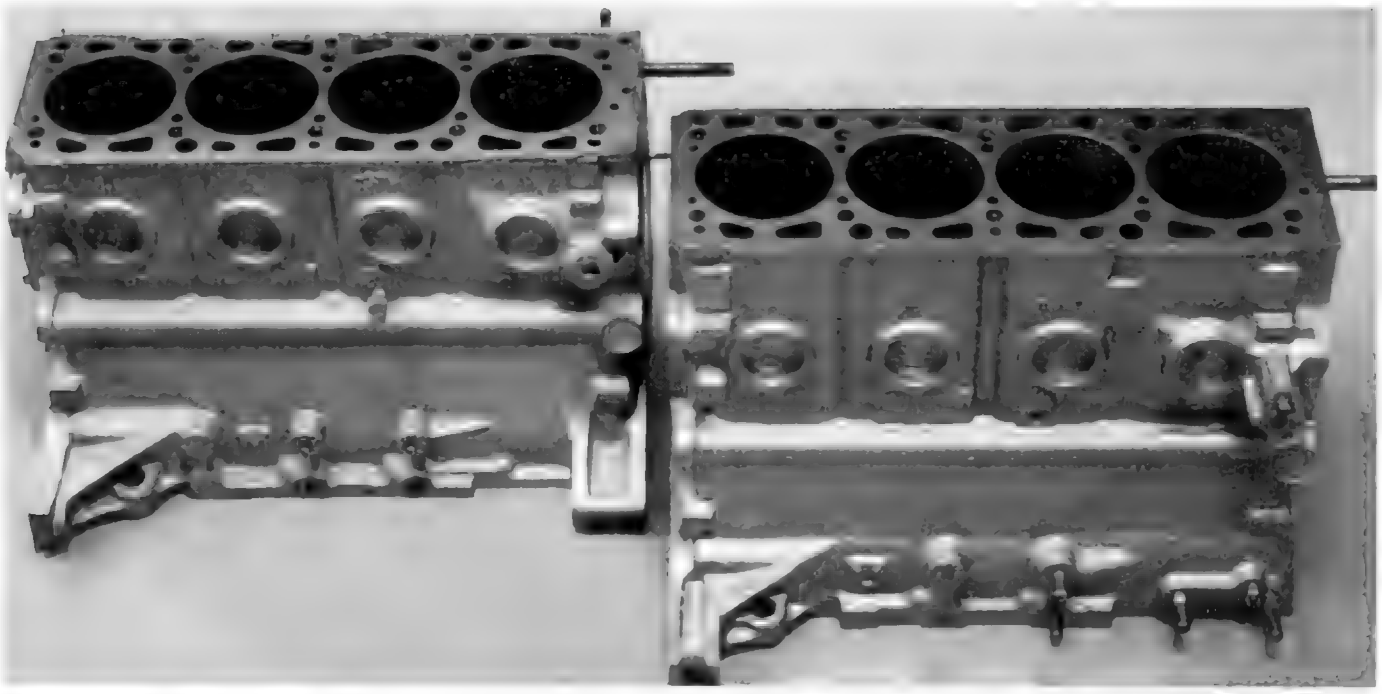
SPEED (rpm)	TORQUE (lbf ft)	POWER (bhp)	BSFC (gm/kW hr)	
5000	123.7	117.8	374	37° adv
5500	139	145.6	306	COMPARED WITH
6000	149.5 max	170.9	292	TEST 4 RESULTS:
6500	145	165.7	288	Better torque 5500 and
7000	145	193.3	289	below, less above
7500	137.8	196.9	322	180 main jet
8000	127.5	194.3	355	190 air corrector



CH3/1: Final test on Tom Casey's 2043cc Fiat engine. Note from this graph and tables on previous page tendency of the engine to run rich around 8000rpm due to its inability to inhale enough air. Note the exponential rise in torque from 5–6000rpm.

CHAPTER 7

BLOCK PREPARATION



7/1: Left, 131 1600; right, Beta 2l, showing different brackets at front of block.

The designs of the various TC blocks display a remarkable degree of commonality. This may be quite surprising in an industry where radical change with every new model is the norm, but it is perfectly logical, and a great advantage to the TC owner as blocks (and ancillaries) can be readily interchanged. Additionally, the owner changing from, say, a 1600 TC to a 2*l* after his first season, will find few surprises in the layout of the larger engine!

The blocks are all cast iron, and no cylinder liners as such are fitted. Early models used an 80mm bore, later ones all 84mm. Cranks cannot realistically be swapped, but the fact that models have a common bore size means that if an owner wants to move up in cubic capacity he simply adopts a longer-stroke model.

Block-mounting points are the same on all models (including the 80mm bore types) which greatly simplifies conversion, although some US blocks (*eg* 1800) are slightly wider than the Euro version. Layouts vary considerably: the Beta block can be used in rear-wheel-drive installation, but the RWD block cannot be used in the Beta because the correct driveshaft mounting points do not exist (although the 124 Sport 1800 block might be persuaded to fit as it has a removable cast alloy alternator bracket in place of the cast iron fixing shown above).

Whereas Beta models were inclined back 20°, later inclined versions (Delta Turbo 1600 *ie*, Thema, Integrale) were inclined forward 20°; the 130 TC and Delta types were mounted vertically. Notwithstanding this, provided appropriate attention is paid to sump design, cam lubrication layout, crank end bearing, etc, there is no reason why an owner considering a conversion should not take this flexibility into account when making his engine selection.

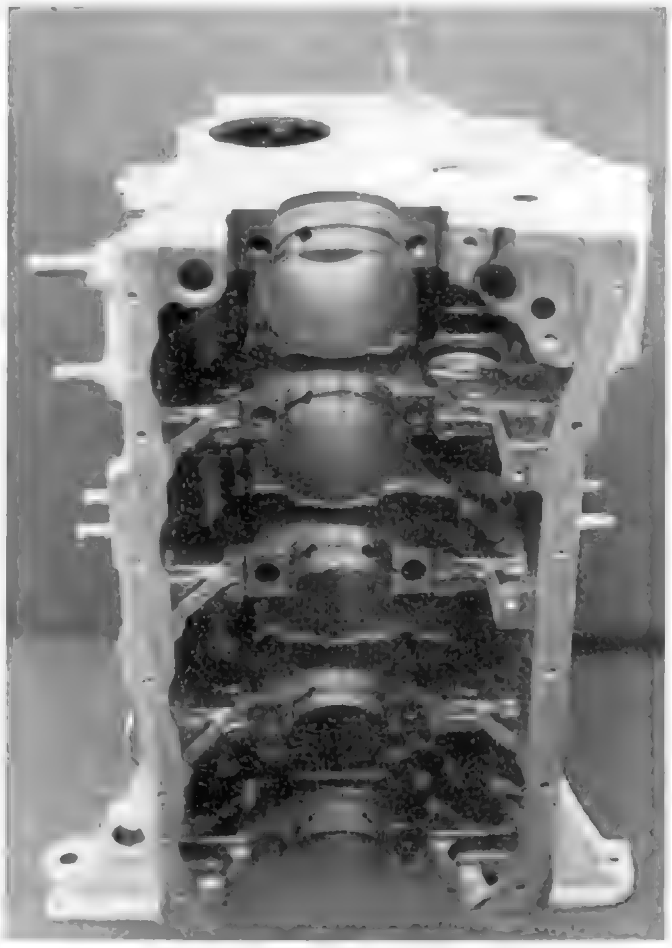
The TC block must rank as one of the classic production models of all time. From modest beginnings with the 1608 model, a cursory inspection of the latest 16v types will reveal that remarkably little has been altered. Adoption from the outset of a five-bearing crank layout and careful attention to the detailed design has resulted in a block of unequalled torsional strength/weight ratio.

Inherent problems encountered on other engines of stress points (and consequent cracking) around coolant galleries and threads are notably absent, and bores can be safely opened out to 2mm oversize without any risk of encountering casting defects in the cylinders. GCT have never had an engine failure which could be attributed to a block defect and have never needed to resort to line-honing a TC block – a time-consuming and expensive process. If in serious doubt (due to bearing damage) measure the housings with a bore gauge.

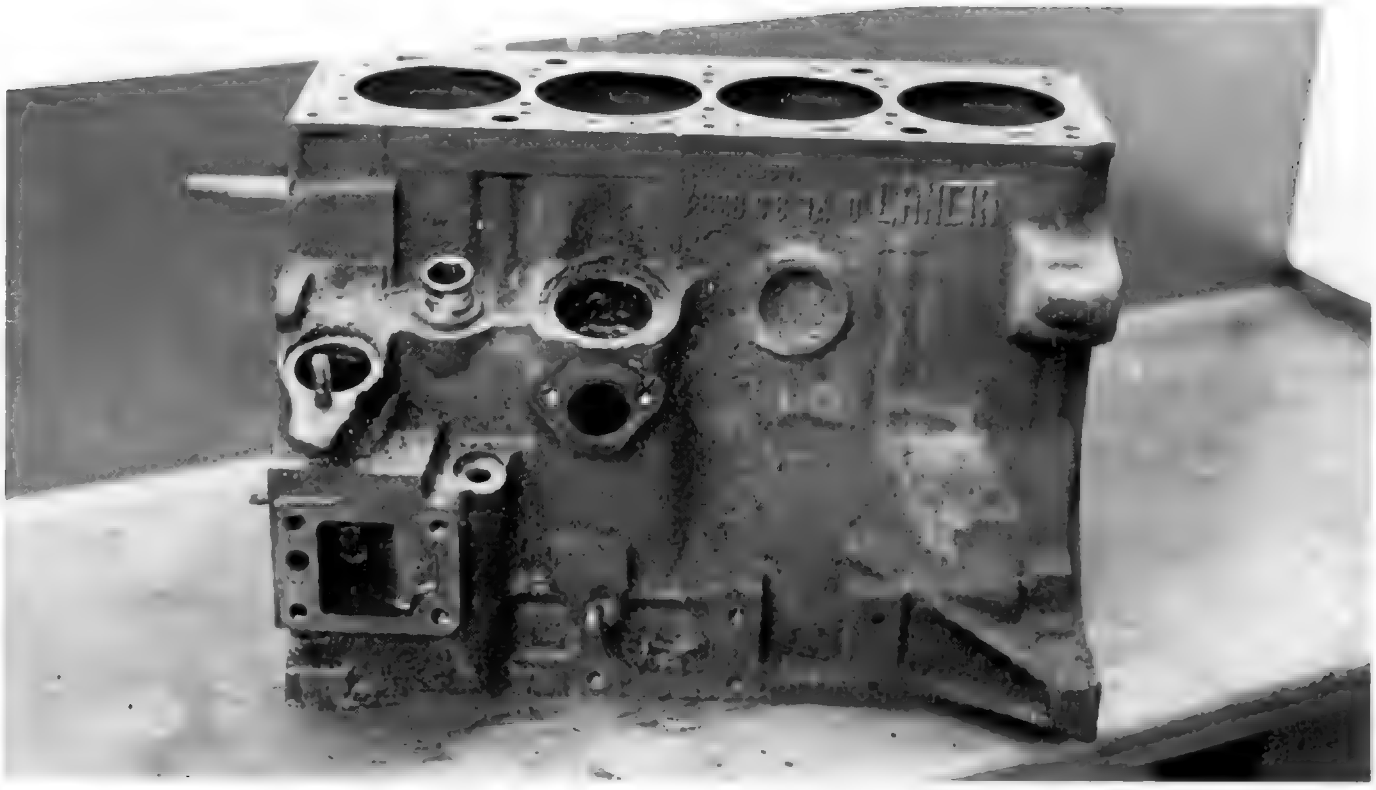
Generally speaking, block preparation at GCT has tended to consist merely of refacing, reboring and cleaning, which is good news for owners, whose budget can then be devoted primarily to other things more viable in the quest for power!

Cleaning – preliminary

Fully strip the block and remove the coolant core plugs, but leave the tensioner stud in place and do not remove the



7/2: Clean, dry and lightly oiled! Journal housings have been lightly dressed with 220-grade emery paper to remove any deposits which might distort bearings. Note driveshaft bracket mountings at top left. This engine is from a transverse layout. Note clever design of reinforcing webs around journals. Despite its immense rigidity, TC block is one of lightest around.



7/3: Lancia Beta 2l block stripped and ready for cleaning. Core plugs not yet removed. Badly rusted coolant galleries can be cleaned with acid – but make sure it doesn't attack auxiliary driveshaft bearings. This is one of the blocks in the first photo of this chapter – before preparation!

REMOVING BALANCE SHAFTS

To remove the balance shafts on late models, first unbolt the toothed pulleys and seal housings. Remove the front circlip on each shaft and tap the shafts with a copper mallet. This will push the rubber-coated bungs out of the back of the block: rear circlips can then be removed and the shafts withdrawn from the front. Remove the bearings (ball-race type) which are a push fit in the housings. Mark the shafts so they go back in their correct place. (On reassembly the bungs can be re-used if a small amount of silicon sealant is used.)

auxiliary driveshaft bearings – these are now almost unobtainable; a replacement block may be needed if they are very severely damaged (unusual).

GCT normally immerse blocks overnight in a special decarbonizing solution; the DIY owner will probably need to use Gunk, Jizer, or Comma Hyperclean and an electric drill with a variety of wire brushes to achieve the same effect. Blocks are normally caked in a mixture of old oil, varnish, rust and road dirt, but as the pictures show, a little hard work yields measurable dividends! After decarbonizing, GCT wash the block in Hyperclean prior to machining.

Boring and honing

The selection process for this is dealt with extensively under *Pistons and Rings*. Once the decision has been made as to which way to go, it is primarily a question of finding someone with the right equipment to carry it out.

Reboring

This may be accomplished with either a boring pillar or a portable machine. Portable reboring tools are perfectly satisfactory, but as they are mounted on the block face, it is worth considering refacing first. A boring pillar (or a milling machine) tends to give better accuracy, but very often a good boring job is spoiled by poor honing – leading to bore ovality and taper.

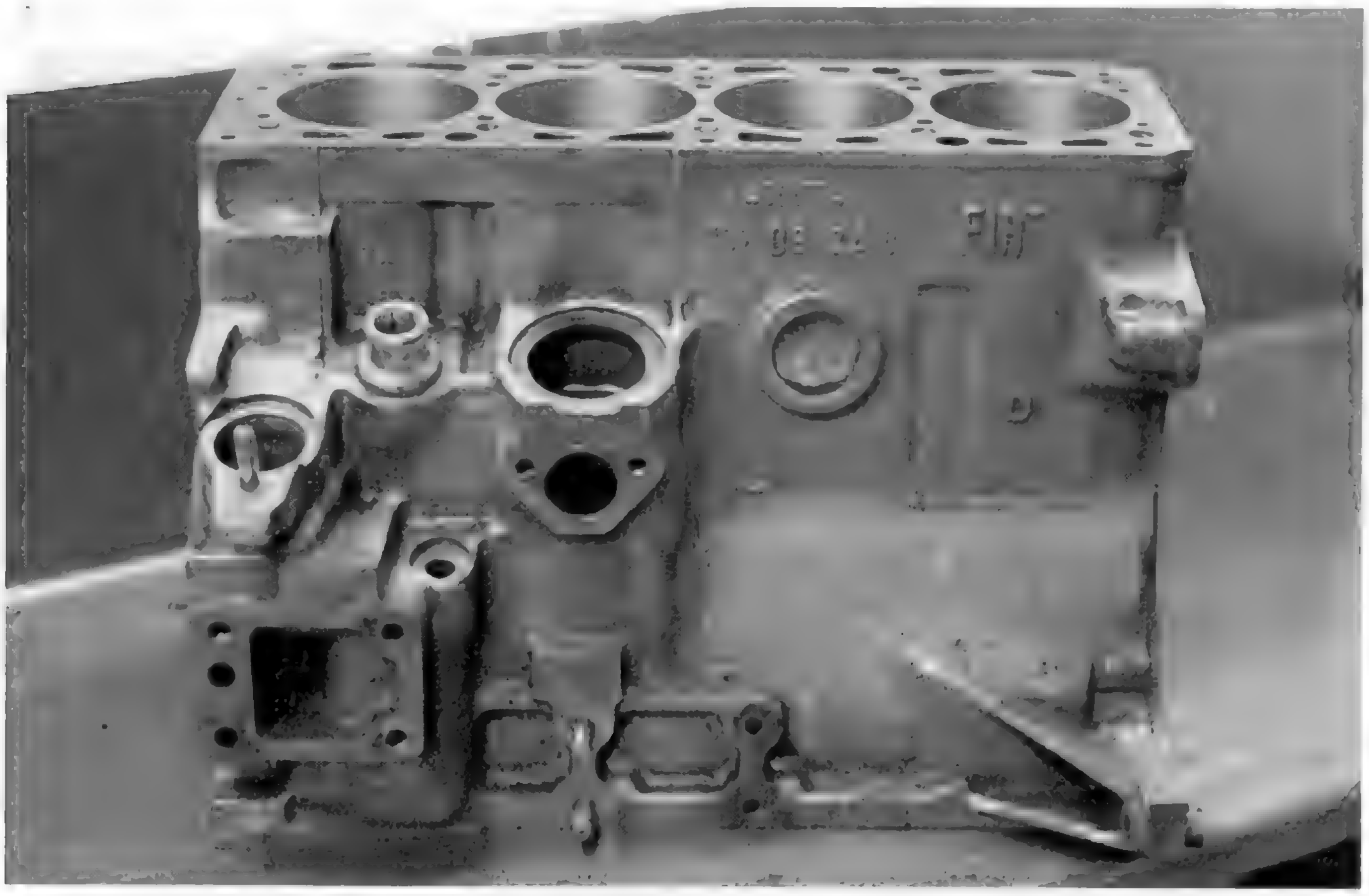
A well-finished bore should have no more than 0.015mm out-of-true in both cases, and ±0.005mm is better. A bore which comes back more than 0.02mm out-of-true should be sent back to be done again to the next size up – although a debate then normally ensues as to whose measuring instruments are actually accurate! (GCT regularly recalibrate micrometers with 'slip gauges' to ensure absolute accuracy. Remember to recalibrate the bore gauge for each block because temperature affects its accuracy.)

Such is the accuracy of production boring that the usual reconditioner's method of centring the reboring machine on an unworn section of the bore (at top or bottom) is perfectly satisfactory – a search for blueprint dimensions is not likely to be successful.

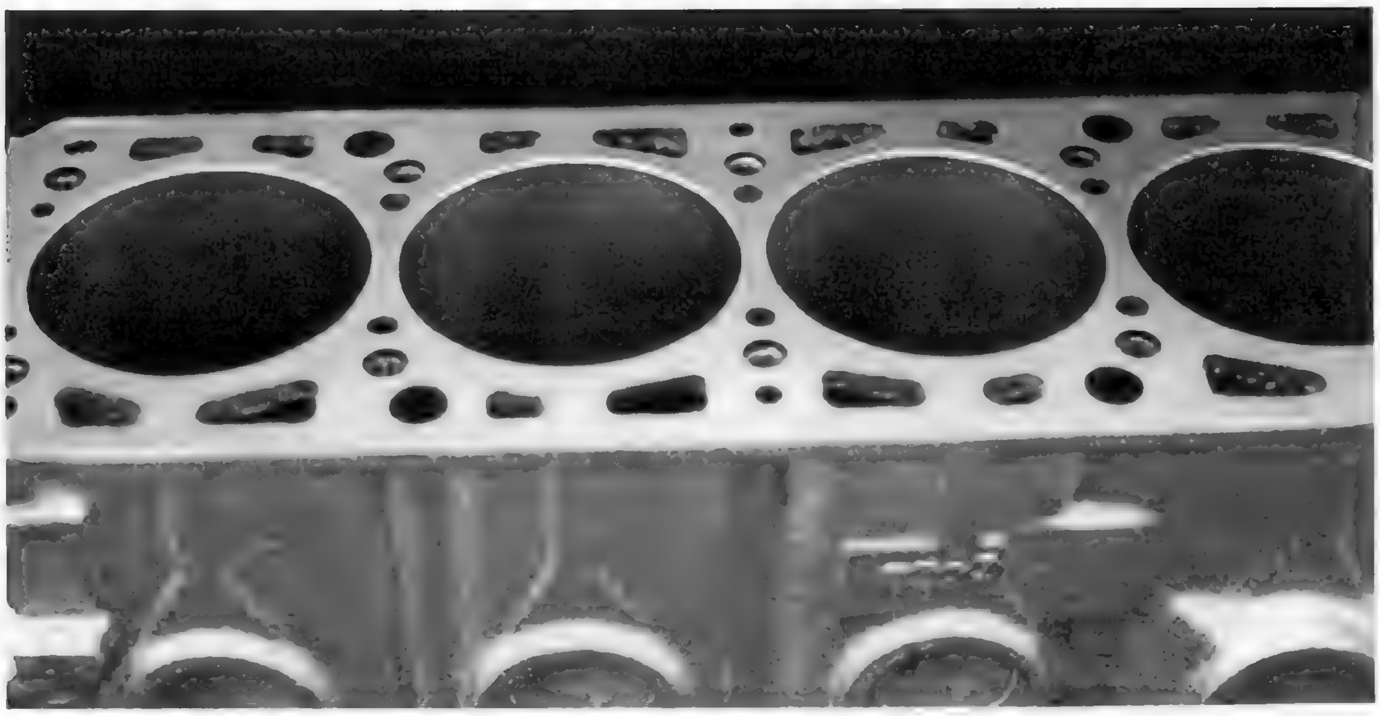
The best reboring operation will machine to within ½–1thou" of the finished size and then complete the job with either a parallel stone honing tool or Flex-Hone. Cross-hatching the bores at 45° (possibly finishing with a few seconds at 30°) is achieved by rotating the hone at low speed (around 250rpm) and simultaneously plunging the tool in-and-out of the bores. Cross-hatching is vital to ensure oil retention.

Normal rebore sizes tend to be 80.4, 80.6, 84.4, 84.6, 84.8. It can be difficult to rebore a block from, for example, 84.4 to 84.6 using portable machining; a boring pillar or mill is better since its greater rigidity ensures that the cutter will penetrate adequately. Normally it is better to go 0.4 larger.

Always source pistons prior to boring because the dome configuration needed may not be available in the particular (first



7/4: Fiat 131 2l block after 24 hours in decarbonizing fluid, detergent wash. Block has been bored and micro-honed and threads cleaned with tap. Treatment next is wire brush and final clean with solvent to remove all traces of oil before painting.



7/5: Shot of refaced block, 'cross-hatching' of hone finish can just be seen. Note larger coolant galleries on exhaust side of block on this early unit. Later models had 'reversed' layout.

[*Author's note:* Every GCT engine since 1987 has been finish-honed with this unique tool. As a tribute to its effectiveness, John Day's St III race Beta, at time of writing, has competed for no less than five years in the Italian Challenge – and still develops 95% of its original dyno bhp – with *no* interim rebore or re-ring! Anyone who has raced against John will know how quick it is!]

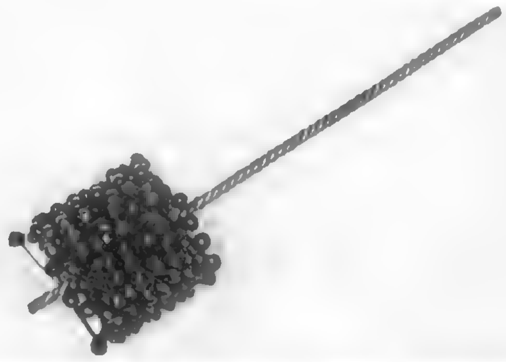
choice) size required. Forged pistons can be made to any size, so availability of ringsets is the main factor when choosing the bore size. Make sure when specifying the required bore size that it allows the correct running clearance for the pistons – if in doubt show them to the operator.

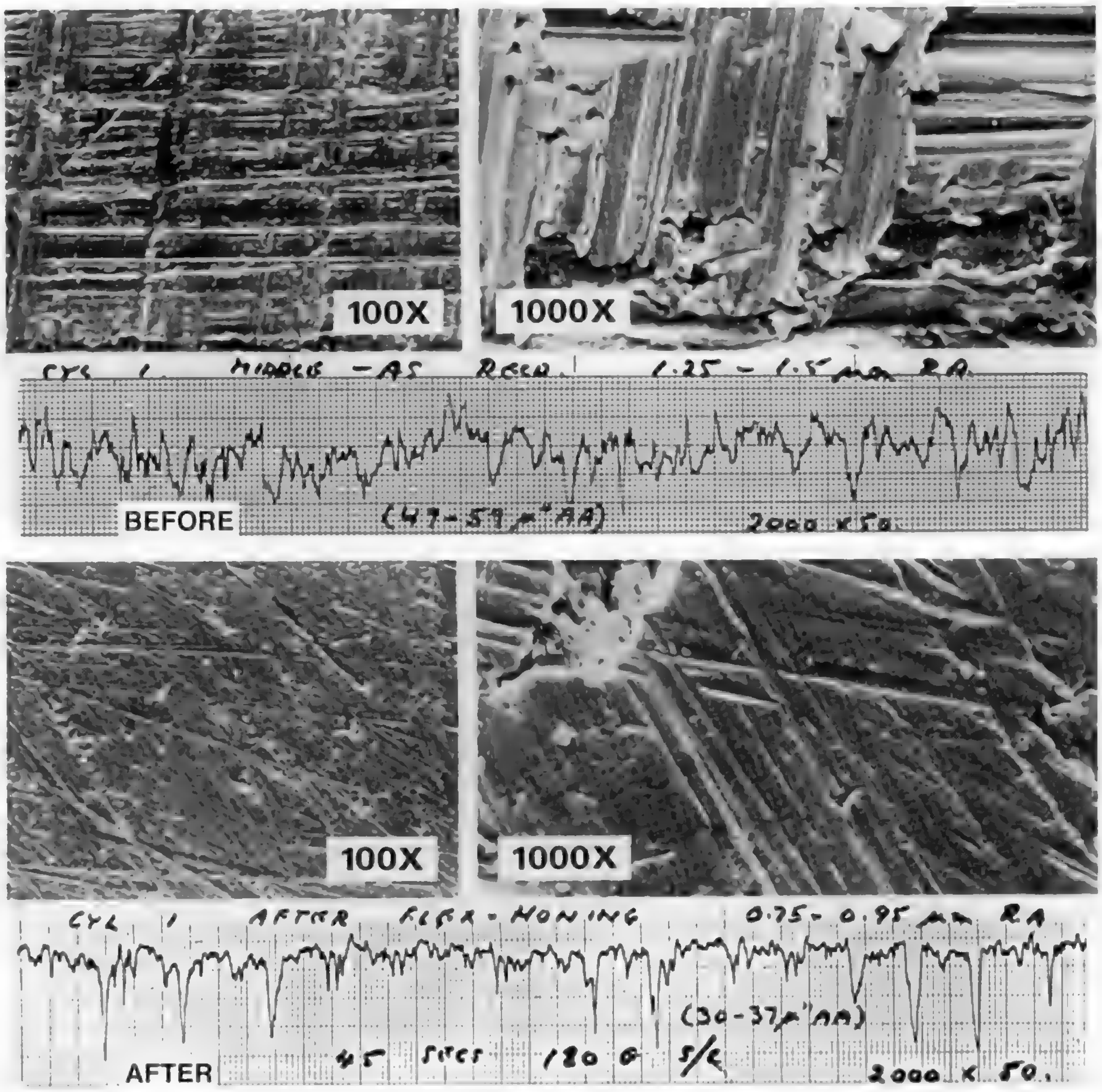
Sleeving

It is perfectly possible to sleeve a TC block where, perhaps, one cylinder has gone over the class size limit, or suffered damage (perhaps due to piston or valve failure). Cast iron liners are available from AE, which can be pressed into a bore – suitably machined – (interference sizes are available from the manufacturer) and then bored and honed to size. Ideally, the old bore should be machined with a step at the bottom of the bore to ensure that the liner can never slip down.

FLEX-HONE

Flex-Hone has been adopted by nearly every engine manufacturer in the world (including Fiat!). Unlike parallel hones, the flexible, self-centring Flex-Hone gently removes jagged edges from the bore surface (visible under a microscope), resulting in a superior finish, which *greatly* extends ring life. Sharp edges and folds/tears in the metal are eliminated and the graphite core is exposed for good oil retention. Various grits and sizes are available, and the tool will last for hundreds of bores. The tool is driven by a heavy-duty drill using special honing oil as a lubricant. Superb technical literature is available from Pacehigh Ltd, of Hatfield, the sole appointed UK and Ireland distributors for Brush Research Manufacturing Co, of Los Angeles.





7/6, 7/7: Flex-Hone brush has silicon carbide (SC) beads mounted on flexible plastic stalks. Pictures from scanning electron microscope and surface roughness graphs show condition of a block (not Fiat!) as received from an engine manufacturer using conventional stones and then after Flex-Honing for 45 seconds with 180 grit SC. The surface finish was improved from 49–59 CLA (peaky and saw-toothed) to 30–37 CLA (plateau). The best Flex-Hone finish with 180 grit SC can be as low as 31 CLA, depending on the original condition. Stones can lead to a finish as rough as 79 CLA – greatly increasing running-in time and reducing ring life. Flex-Honing time is normally 30–45sec. Increasing the rate of up-and-down feed with the drill for the last 5–10sec gives a final superposition of 30deg cross-hatch on the 45deg finish. It is possible, with accurate boring equipment, to bore to within 0.5thou" and finish hone entirely with the appropriate grit Flex-Hone. [Author's note: Interestingly, one of the endorsements in the Flex-Hone information cites the case of a manufacturer of WW2-vintage aero-engines: Flex-Honing for 20–30sec gave a ring-seating time of 30–60 minutes compared with 20–30 hours previously!]

Honing

Some companies use large-capacity honing machines to carry out what is effectively a reboring operation, but this is only really possible with a sophisticated machine; honing after boring or in preparation for re-ringing is usually carried out with the hone driven by a heavy-duty drill.

For finish honing, either to re-ring or post-rebore, the best tool is definitely a Flex-Hone, although less sophisticated 'glaze-busters' are available. In the case of re-ringing the objective is merely to remove the varnish 'glazed' to the bores to open out the pores of the metal and restore the cross-hatch. The correct grit for honing (including piston-rebore) is 180 grit for chromed top rings and 220 for cast iron/moly-faced types. A fine finish is not always a good sign.

Refacing

This operation can be carried out on a large-capacity milling machine or surface grinder.

GCT employ an 8" diameter single-point flycutter fitted with a carbide tip – tool steel is too soft – with the milling head tilted approx 20min of angle so that the tool leading edge will give a clean final cut. This operation can quite satisfactorily be carried out with the block mounted on the skirt. Never remove more than a 'minimum skim' for flatness unless the block has been dry-built *(see later)* as the piston set-up height may be seriously upset and, remember, as with the cylinder head, removal of material will reduce the valve-piston radial clearance by approximately ⅓ of that amount.

GCT have machined as much as 3mm off TC blocks, but such high removal can cause problems – cams may not 'dial-in' without the use of adjustable, or vernier pulleys because the distance from the inlet cam pulley to the crank is radically altered, and there may not be sufficient adjustment in the tensioner pulley to take up the extra slack in the belt. If removing a large amount, make sure that the head bolts do not bottom-out – they may need to be shortened – also ensure that the gasket locating dowel holes are deepened. After refacing, chamfer the bores to ease the fitting of the pistons and rings – they won't go in if this is not carried out, or at best will be damaged. A blended ½mm × 45° chamfer is fine.

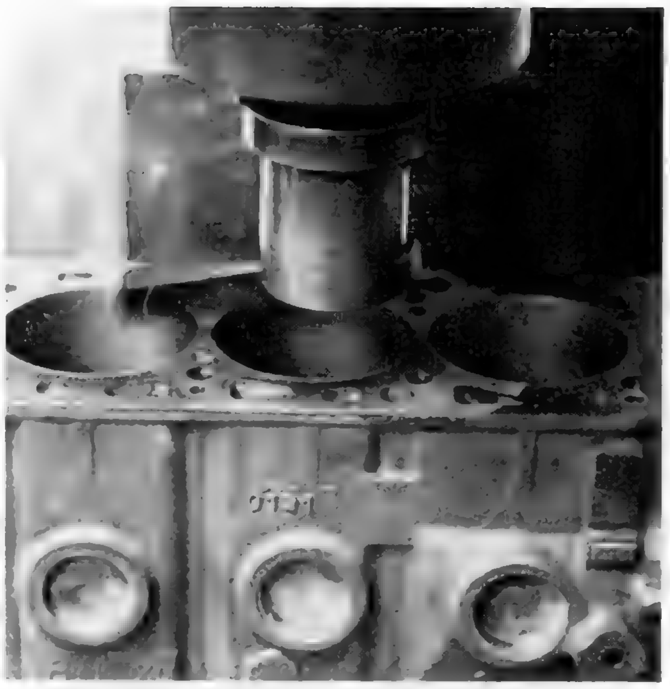
It is important to achieve a fine finish on the block face, although the tiny grooves from machining and grinding help to 'key' the gasket in position. If the grooves are too coarse, the fire ring of the gasket may not compress adequately and gas leakage may result.

GCT have only ever encountered two (out of many hundreds) cracked TC blocks. Cracks will show up readily after refacing.

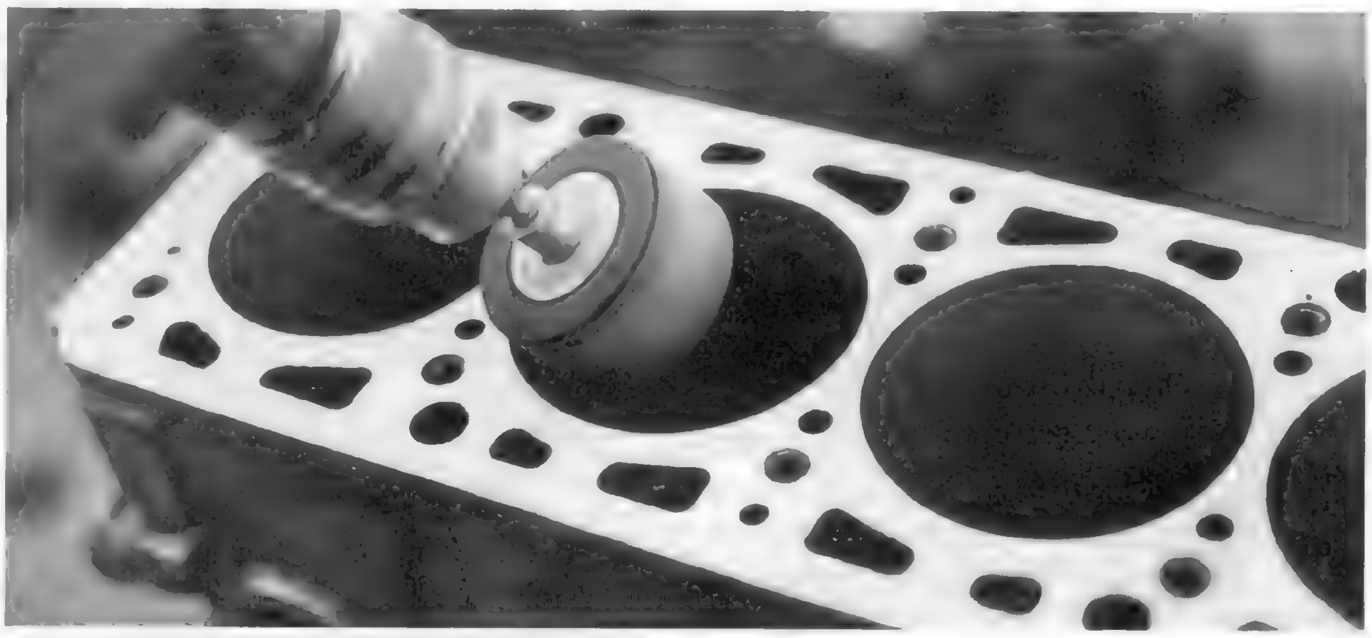
In one case, the crack was between two adjacent coolant galleries and an oil feedway on the exhaust side; the other was between two adjacent bores, where the land had been overheated by a blown gasket. If there is any question as to whether a crack is present, polish the surface with 220 grit and examine with a magnifying glass. Such cracks are not repairable and a replacement block should be sought.

Wire-ring/Wills ring conversion

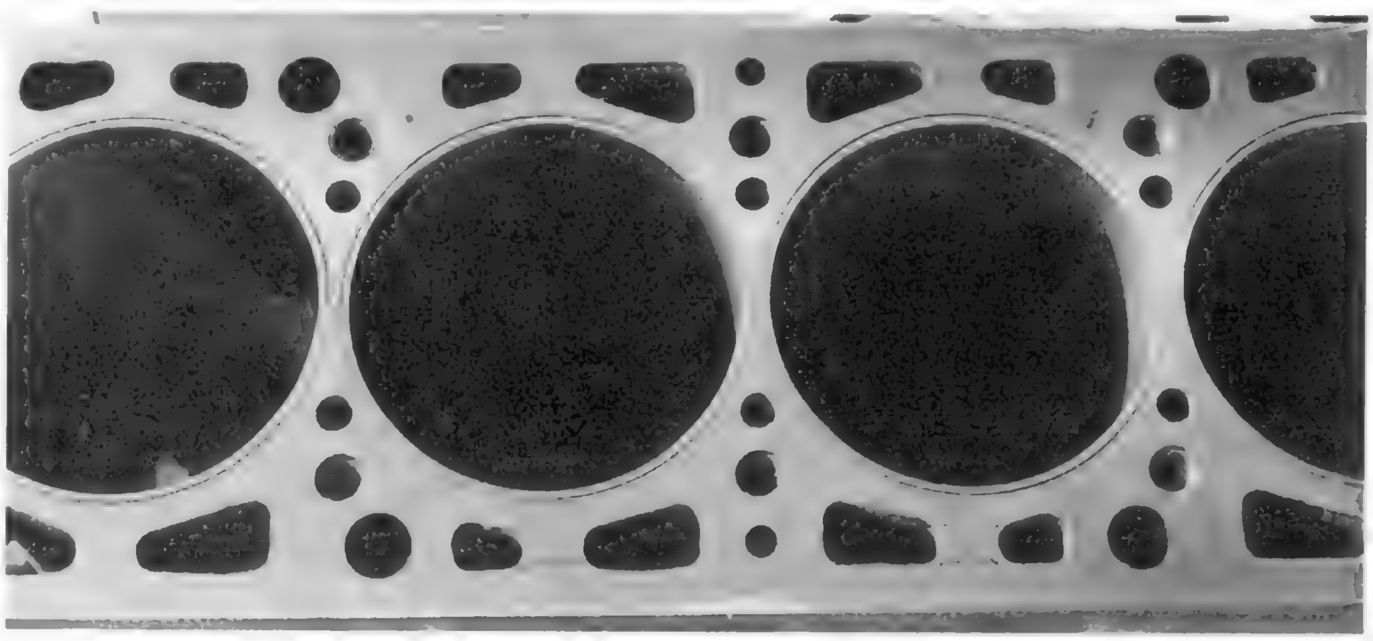
Conventional gaskets contain 'jacking points' to raise the clamp load in key areas – *eg* around bores and oilways. This clamping load can be augmented by the use of a wire ring insert under the centre of the fire ring.



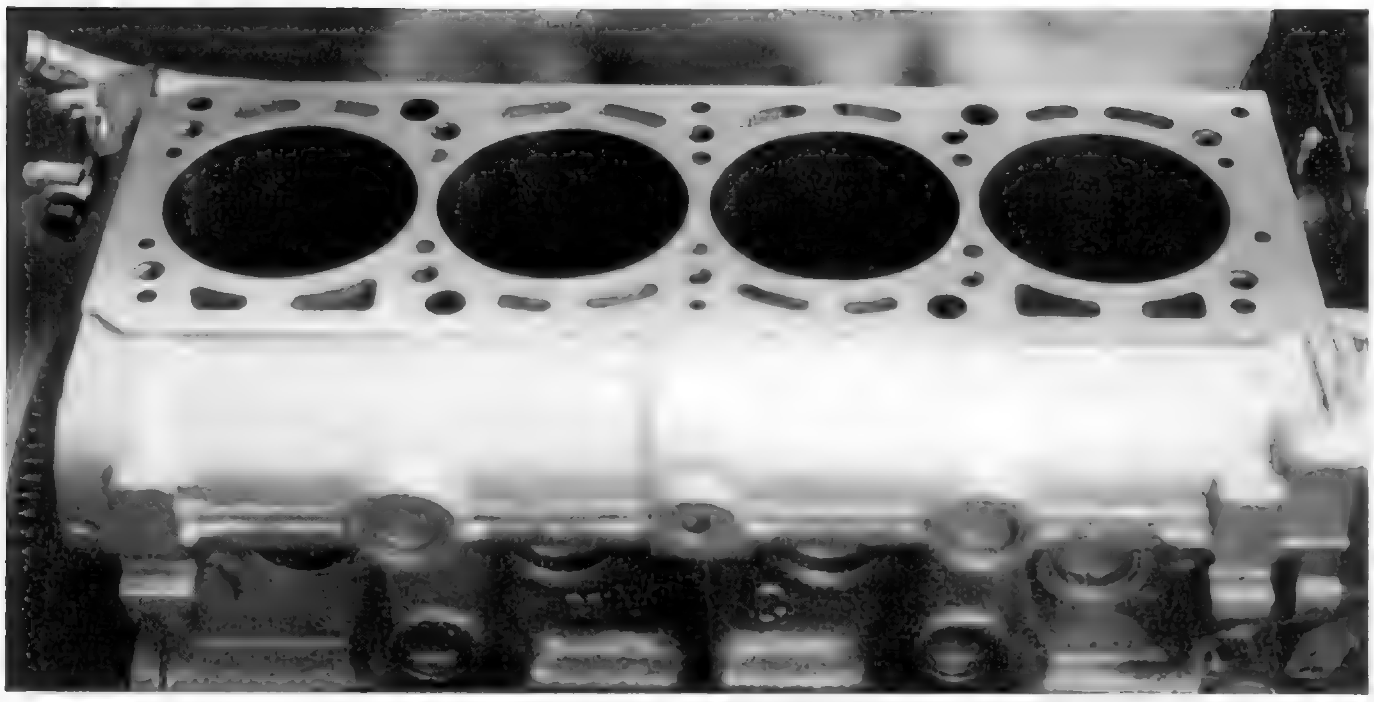
7/8: Refacing block on mill using carbide-tipped flycutter; 3–5thou" usually cleans a good used block, but a severely overheated block may need 10–15thou". Refacing should be carried out prior to rebore if 'portable' boring tool (eg Van Norman) is being used as tool locates on blockface. Cutter speed is 570rpm, bed feed 120mm/min.



7/9: This technique was criticized in some quarters when the book Fiat and Lancia Twin Cams was published. Item is 220-grade carborundum drum sander (ATA) which is being used to chamfer tops of bores; final dressing is by hand. Chamfer is essential if piston ring damage during assembly is to be avoided. This quick, simple technique has been used on hundreds of GC-prepared engines. Doubtless, a more expensive and complicated method could be devised!



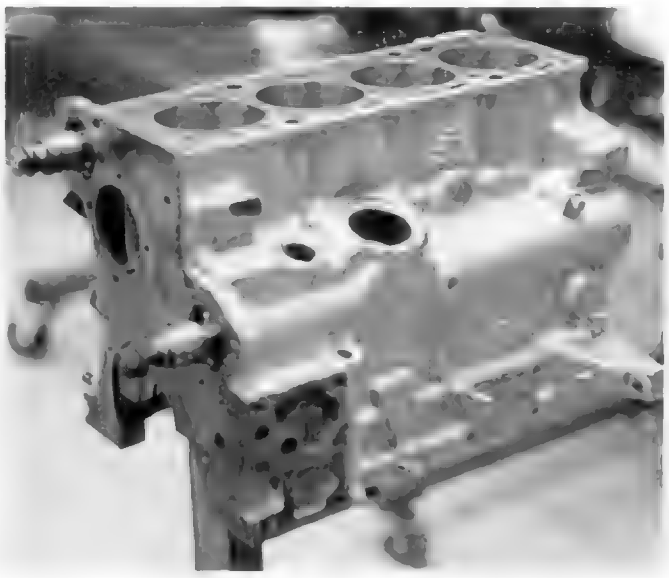
7/10: Wire ring conversion: small grooves machined around bores hold copper wire to augment seal of copper gasket. Grooves must be carefully sized to 'nip' wire without undue force that might crack edge of block. Proximity of bores makes 'Wills ring' conversion on TC very difficult on oversize bores.



7/11: Gp A 8v Integrale block being built up. Extra bulk and weight of balancer shaft assemblies readily visible.

Special tools are available to cut such grooves, or the job can be carried out using a mill and small single-point tool. The groove width must be dimensioned to nip the wire in place. Stainless steel, steel, silver or copper wire can be used, but in GCT's experience the wire ends should be welded, soldered or overleaved to prevent leakage. This technique is only needed on high-boost (20lbf/in^2+) turbos as the latest OE gaskets from Fiat are perfectly strong enough up to this limit if used with race bolts. The wire, fitted, should protrude a maximum of 2thou" for stainless wire and 4thou" for silver/copper.

Wills ring conversions should not be attempted on late engines with bore sizes over 84.4mm since the smallest rings (which are gas-filled) available are $\frac{1}{16}$"dia, and insufficient land would remain between bores. The ideal method of location is in a clearance groove (data is available from the manufacturers) but a plain counterbore is easier, though care must be taken to ensure that the compressed rings do not intrude onto the pistons when the head is fitted. The remainder of the block is sealed with a paper gasket, which must be carefully cut to fit – very time-consuming – and nitrile rubber O-rings around the pressure oilways. (Note that Wills rings must be retorqued *cold* before power runs.) Forget wire/Wills ring conversions on normally aspirated engines – if the 130 TC gasket blows, the gasket *per se* is *not* to blame (detonation or overheating, defective bolts or warped head/block faces are the main culprits).



7/12: Engine has crank nose-driven oil pump, hence recess in block for pump housing. Similarity to earlier TCs is not just skin deep; many parts are directly interchangeable.

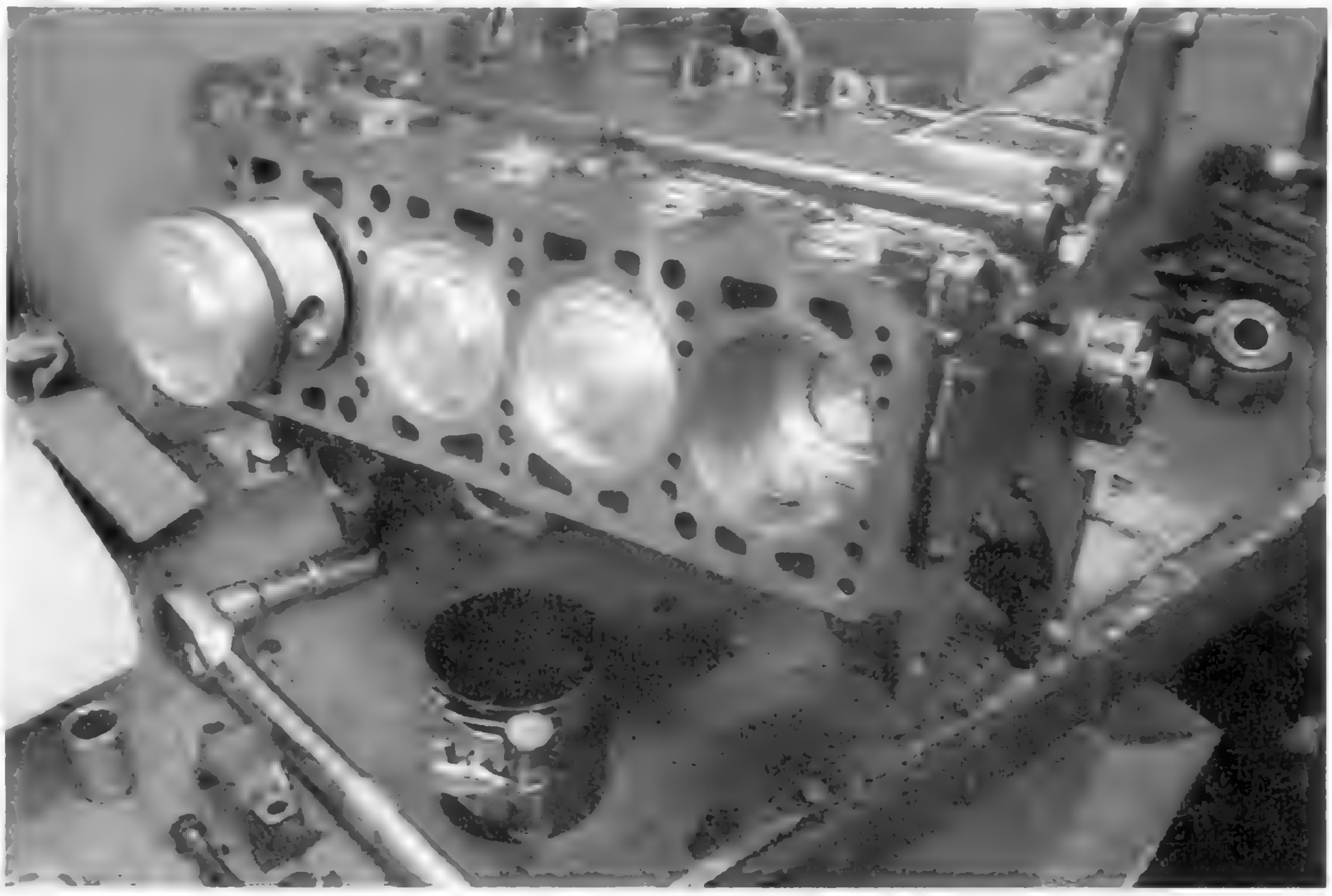
Final cleaning

Start by descaling the block with a wire brush and electric drill – this will give a good key for paint. Then clean the head bolt threads (M10 × 1.25) and sump bolt threads (M6) with taps. Clean the crank journal housings, end faces, etc, with 220 grit and oil. GCT normally chamfer the various oilways and head bolt threads in the block face as well. A good scrub with an engine cleaning brush and Hyperclean, for example, is then followed with a hot detergent pressure-wash, water rinse and air dry.

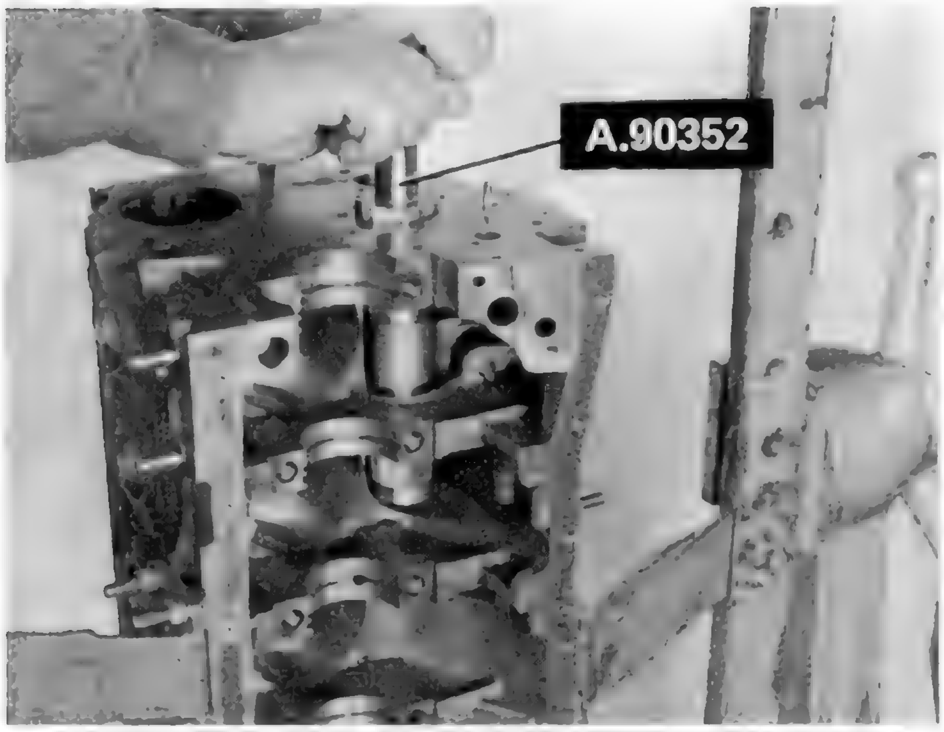
All machined surfaces must be immediately oiled – WD40 is ideal and indeed this can be used to 'chase' water off the block. It is not worth removing the oilway plugs (they may leak when new ones are fitted) as a high-pressure nozzle will clean them quite satisfactorily. Washing the bores with hot detergent is most important after boring/honing. Carborundum dust lodges in the pores of the bores and use of solvents drives it even deeper. Only detergent will remove it – if it is left in place, ring life will be very short.

Make sure that all water is blown out of oilways and threads (eye protection should be worn in case any grit is fired out by the airline) and then, oil and wipe the bores with paper until all traces of grey honing dust are removed. Only at this stage can trichloroethylene – 'trike' (or derivative) spray be used to final-clean the bores, but oil them afterwards.

If at all possible, leave the block overnight to air-dry, then spray the cast surfaces of the outside of the block with 'trike' or brake cleaning solvent to remove all traces of grease/oil. GCT paint the block with Hammerite, which is an



7/13: 1600 Fiat block prepared by Gareth Jones shows very high standards achievable 'at home' by DIY owner. Pistons are oversize 131 1600 (9.0:1 CR). Hundreds of Fiat TC 'clubman racers' have followed same route as Gareth, built their own engines (often via a long-distance 'correspondence course' from GCT!) and been delighted with results.



7/14: Extract from 130 TC manual. Replacement of the auxiliary driveshaft bushes requires use of a special reamer – if they don't look too bad, leave alone. (Fiat Auto SpA – copyright reserved)

outstandingly durable device for making a block look like new! Do not paint the end faces of the block or machined faces such as the oil filter housing mounting point as the paint may interfere with gaskets; clean them and the core plug bores with 220 grit and scrape off any traces of old gaskets with a sharp blade.

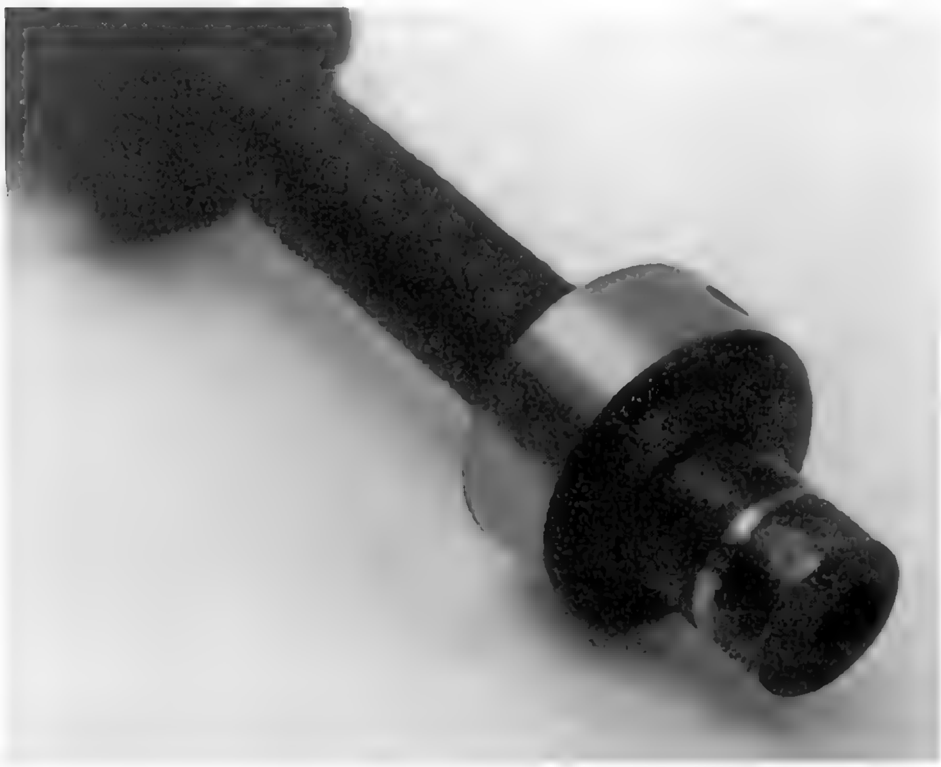
Thorough cleaning is really the key to good preparation, and GCT usually spray 'trike' through all oil galleries and head bolt threads, followed with a good blast with an airline just prior to the build-up.

Auxiliary driveshaft

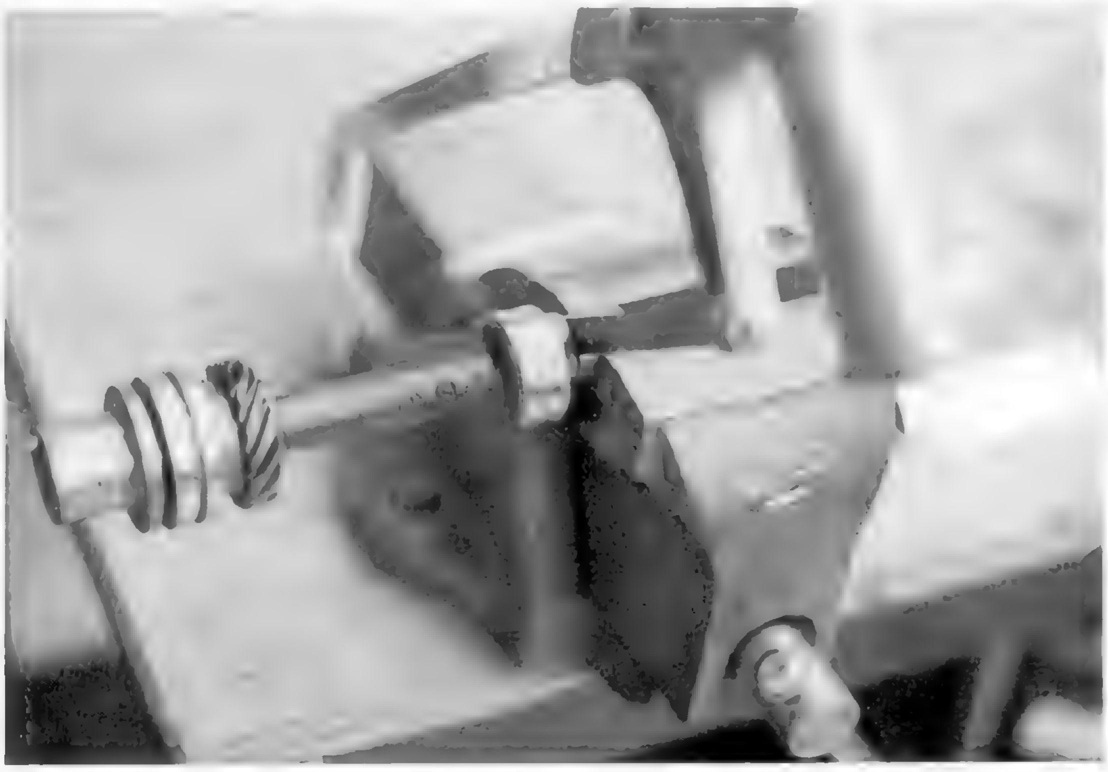
Dress out minor damage to the auxiliary driveshaft bearings with 1000-grade. They are not heavily loaded so do not attempt to replace unless (a) the bearings and (b) the special fitting and reaming tools are available.

Early engines of 79.2mm to 90mm stroke suffer from an inherent problem that if the auxiliary driveshaft is not correctly 'timed-up' when the cam belt is fitted, the fuel pump lobe can hit No 2 con-rod.

Dress up the seal and bearing journals of the auxiliary driveshaft with 1000-grade and clean out the oil gallery with solvent and an air line. As with most engine internal parts, oil the shaft if it is going to be stored for some time.



7/15: This one shows strike mark from conrod contact. Best thing to do is...



7/16: ...cut it off (a lathe is not needed – the shaft merely has to clear the rod safely!)...

7/17: ...then plug oil gallery by...

7/18: ...drilling out to 9mm, tapping...

7/19:...and fitting ⅛ BSPT plug. Do not put it in too deep, or it will block oil feed to rear journal. Use Loctite Hydraulic Lock on threads. (Note: Most models will accept ⅛ BSPT plug, but check size before modifying.)

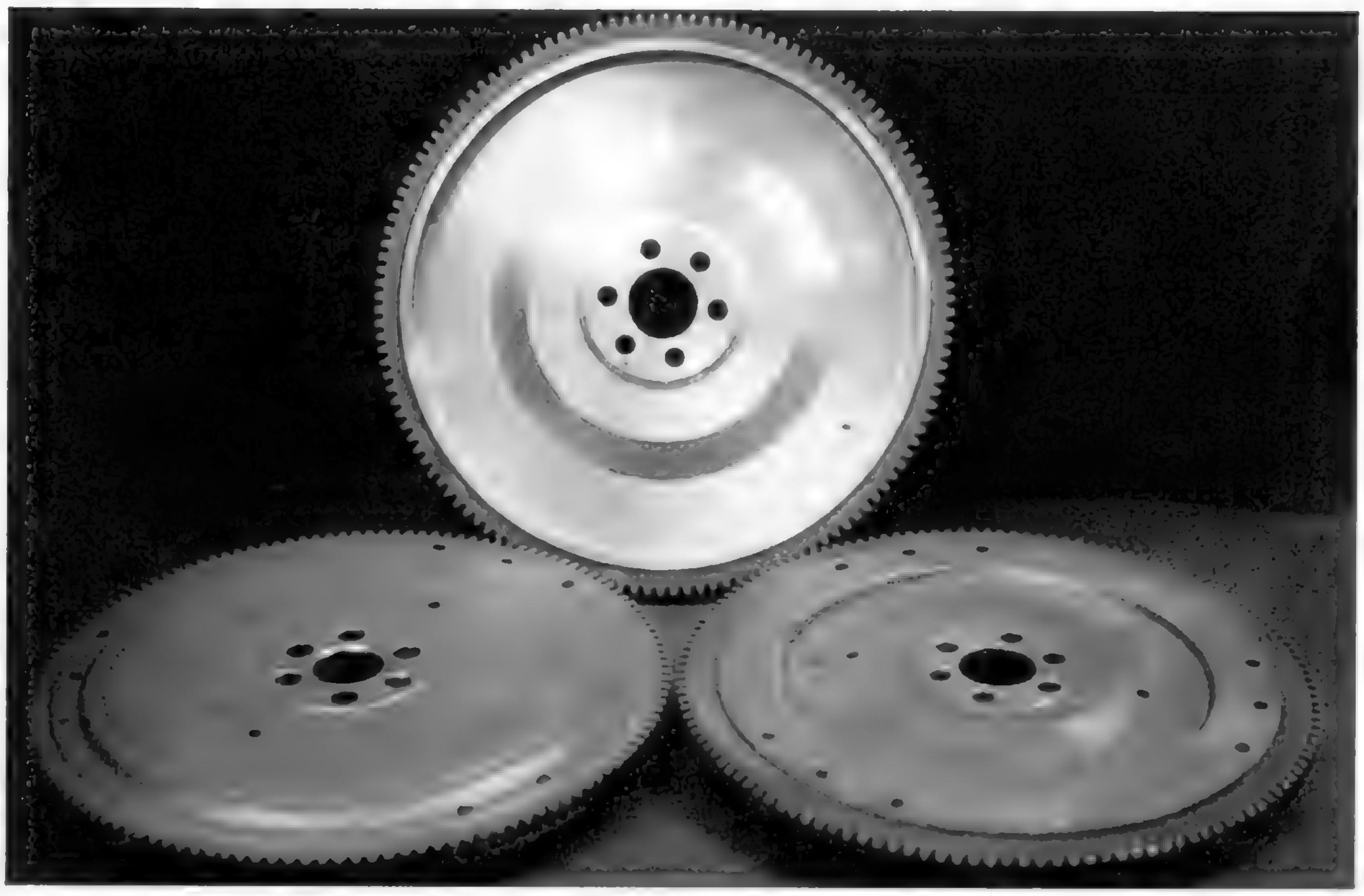
CHAPTER 8

FLYWHEEL, CRANK AND ROD PREPARATION

Materials

Crankshafts: 1585 engine – cast iron; 1592, 1608 and 1756 engines – forged steel; 1995 engine – heat-treated forged steel (nitrided or Tufftrided).
Con-rods: All engines – steel forgings.
Flywheels: All models except 2*l* turbo – cast iron.
[*Author's note:* Homologation papers for the Integrale state that the flywheel is steel. Machining work on these at GCT would indicate that if it is steel, it is made from cast steel rather than, say, rolled En8 plate. In the absence of any hard data I would suggest treat as cast-iron for lightening purposes.] (8/1 – 8/3)

8/1: Top to bottom: 1585 131/Beta, 1756 124 Sport (longer nose than 132 1800 and 10mm dia flywheel bolts rather than 12mm) and 1995 (2l) 131/132/Beta. 1585 crank is cast, other two are forged steel, as are 1592, 1608 and all other 2l models, eg Integrale. Cranks are exceptionally strong.



8/2: Flywheels: left – standard cast iron 2l, centre – medium-weight steel 1600, right – standard 131/Beta 1585. Note on steel item that thickness around centre of flywheel must be retained as shown. On steel flywheel either secure ring gear to flywheel with grub screws or weld (balancing is essential). Note 2l has 12mm bolt holes, 1585 10mm; 1585 can be opened up to fit 2l but has smaller friction face (except Delta 1600 Turbo). Late 1756 (Fiat 132) has 12mm bolts; if using on early 1756 (Fiat 124) sleeves need to be fitted around 10mm 124 flywheel bolts. Step on flange at centre causes stress raiser on standard flywheel. On steel items it must be radiused as shown – better to leave it out altogether.

Crack-testing

GCT have only twice seen a cracked TC crank, in both cases 2*l* items where minor heat cracks had been caused by bearing damage, but always inspect the crank journals prior to grinding. A crank which has suffered chronic bearing failure could have minor heat stress cracks on the journals which may well 'grind out'. Cracks on the journal fillet are more serious since a crack here can lead to crankshaft breakage. If in doubt, the crank should be crack-tested by Magnaflux or a similar method where the crank is coated in magnetic dye which shows up cracks under ultra-violet light when the crank is magnetized. (Use of a powerful magnifying glass is a useful DIY substitute.) The most probable causes of cracking are flywheel imbalance (though this usually leads to bolt failure first), chronic bearing failure, or the flywheel hitting the kerb (on single-seaters).

Similarly, flywheels should be carefully inspected at least once each season; cast iron flywheels are liable to crack where the bolt flange joins the friction face due

8/3: Poor-quality grind, use of previously worn crank (note eccentricity of fillet radius), excessively heavy flywheel and torsional stress led to failure of this hydroplane 1800 crank. Driver Neil Allen reported "I switched off the engine coming into the pits, but cut it a bit fine and couldn't restart it. Then I looked over my shoulder and saw the flywheel lying in the bottom of the boat! Just as well it didn't shear out on the course or it would have gone through the hull like a buzz saw!"

to the pressure from the clutch, especially if a competition cover with a high clamp load is used. Steel flywheels have many times the fatigue life of cast iron; both, of course, need to be properly lightened (or designed in the case of steel) to ensure that there is sufficient strength and no concentration of stress.

Heat cracks on the flywheel face very often 'grind out', but the safest solution is to inspect after grinding the friction face and discard as appropriate. Such damage is caused by clutch slip – either because its torque capacity is too low or the actuating arm adjustment is incorrect. Scoring is caused by clutch plate wear – down to the rivets!

Cracks in con-rods are as rare as those in cranks. The standard TC rod (new, or from a low-mileage, well-maintained engine) has a *very* long fatigue life. Owners are highly unlikely to encounter rod cracks during normal racing *in any level of tune* provided the following rev limits are adhered to: 71.5mm stroke – 9500rpm; 79.2mm stroke – 9000rpm; 90.0mm stroke – 8600rpm, and the rod maintenance as given in the 'life schedules' is followed. (8/4)

Naturally, a rod will crack and break (all too quickly!) if the bearing seizes. This will usually be due to a lubrication failure where the assembly has run dry and friction between the bearing and crank has led to almost instantaneous overheating and friction welding between the rod, bearing (or what remains of it) and crank. Expansion due to overheating adds to the local mayhem and in extreme cases rod failure is virtually inevitable. If the rods used for the rebuild do not show signs of distress *ie* 'blueing' from overheating, or rod-crank damage on the rod cheeks, it may safely be assumed that the rods will be crack-free. The rough finish of the standard rods inhibits easy Magnaflux crack-testing around the main stress area at the root of the rod shank and pre-polishing is desirable.

[*Author's note:* I have been unable to establish whether the rough finish on all the TC rods is due to shot-peening at the factory (in which case the shot is very coarse!) or merely due to forging.]

RECONDITIONING ACTION

The following remedial work may be carried out on the crank:

Excessive 'bow'

This important check may be measured by placing the crank on V-blocks (on No 1 and No 5 main journals) and measuring the deflection of the centre main bearing

8/4: Broken con-rod from 1800 hydroplane engine. Owner said "I've got this oil which is the same type as you supplied (it wasn't) but cheaper...." QED: you get what you pay for. This kind of failure is often attributed to the con-rod bolts. It was nothing to do with the bolts: the bearing simply 'let go', and the mayhem shown followed naturally.

with a dti. The factory allowance is ±0.0008". Remedial action can be carried out by pressing (other than cast cranks) on a crank press (most heat-treatment plants have such an item), or by regrinding. Clearly a 'bent' crankshaft can be ground straight, but the out-of-true of the flywheel-mating flange may be significant. The factory figure for this (with the dti placed approximately 40mm from the crank axis) is ±0.001". In practice it is a difficult grinding operation to correct this, and GCT have tended to rely on accurate balancing of the crank/flywheel assembly to ensure that stresses set up from the resulting 'wobble' on the flywheel are minimized. If the crank 'bow' is excessive the crank will be hard to rotate on the dry-build.

Worn or damaged journals

If the journals meet the specified sizes given in the table, only a light polish with 400-grade carborundum paper is required. This procedure (also called lapping) may be carried out in a lathe, with the crank turning in the direction of rotation so that the grain of the journals is not raised against the bearing. Crank grinders normally use special tongs for this; if using a lathe, employ a strip of cloth-backed abrasive approx 3ft long by ½" wide, rotate the crank at around 60rpm, oil the journals and hold the abrasive tape (with the ends well separated – if they are allowed to touch, they may snatch) firmly around them until a consistent matt finish appears. This method is useful for final dressing of a good, unworn crank which has had the oilways modified, where there is a risk of high spots damaging the bearings.

According to the condition of the journals, the decision whether or not to grind the crank can be made. Oilway mods should ideally be carried out first (see photo) and also stress relieving if final hardening by Tufftriding or nitriding is being considered. Tufftriding can be carried out on all TC cranks, nitriding only on 2*l* versions. At the same time, the crank grinder can dress the thrust faces if required and finally lap the front and rear seal faces.

Always specify the size required – ideally, the finished size should fall halfway between the upper and lower limits, but don't expect unreasonable standards of accuracy. The fit between the crank journals and bearings is around 2thou", so ±²⁄₁₀thou" either side of the specified journal sizes is not significant. The reproduction of the fillet radius as per the diagram is important to minimize the concentration of stress where the crank journals and webs meet. (8/5)

Crank grinding data (direction, surface, finish) is well summarized in Vandervell bearing manuals – but reputable grinders will know this anyway.

Crankshaft – useful specifications:

Crankshaft end float (with thrust washers installed) all TCs – 2–12thou" (GCT normally build to 4–6thou")

Max misalignment of big-end journals to mains – 10thou"

Max out-of-round of journals and taper – 0.0002"

Crank grinding data – see table (source Vandervell manual)

Crank heat treatment

Tufftriding greatly increases the durability of TC cranks. It is a process which produces a layer of around 12–16μm (approx 1/2thou") in the surface of the crank. This layer contains a high nitrogen/iron/carbon content and gives exceptional sliding (low coefficient of friction), wear and hardness properties and does *not* require use of En40B nitriding steel. Because of this, it has been adopted by many manufacturers using lower grades.

The wear behaviour in fact is more significant than the hardness with this process, and its characteristic comes from the fact that the 'compound layer' is non-metallic and has a reduced tendency to seize and weld with the pairing materials (*ie* bearings).

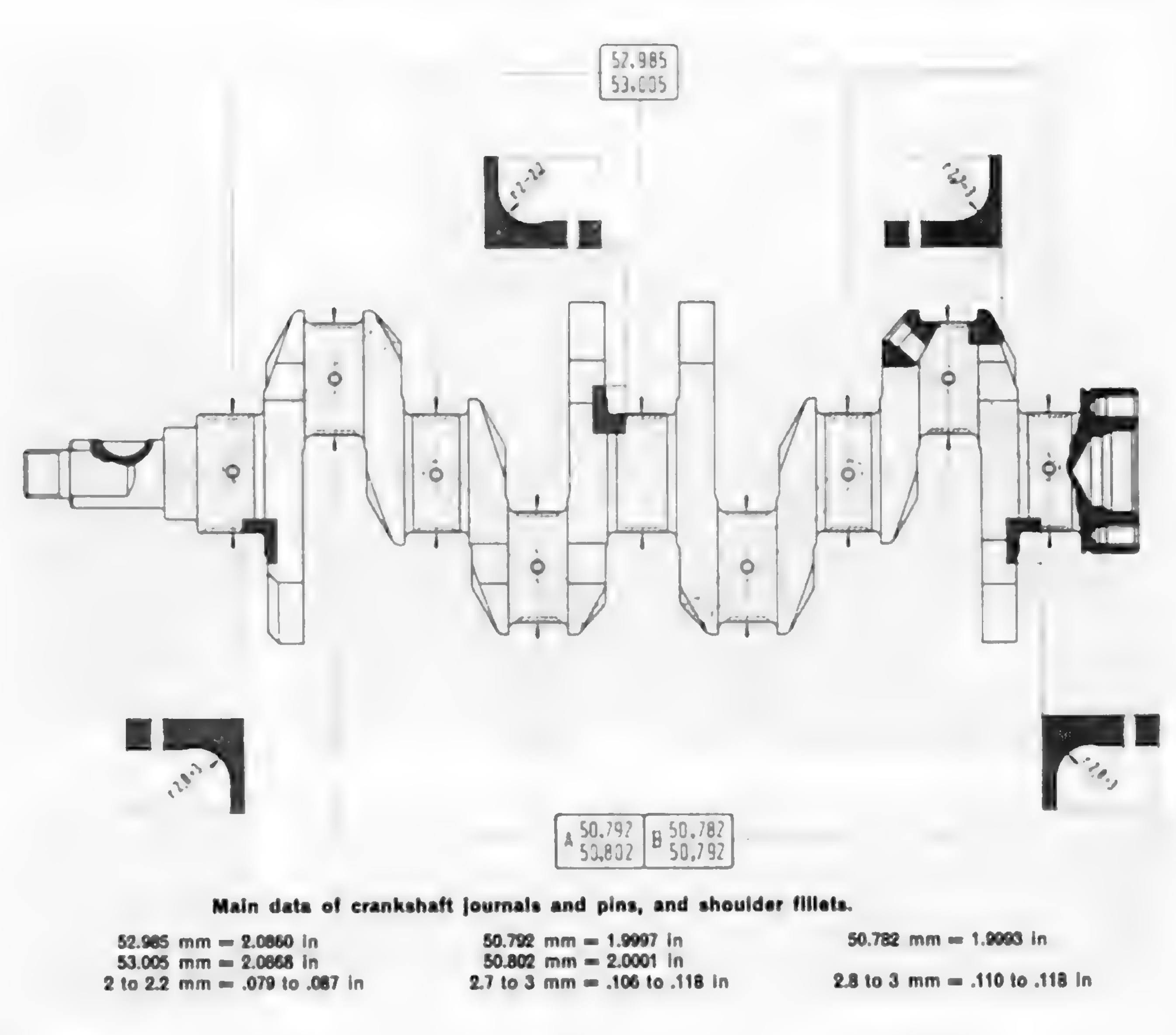
A crank subjected to this process must be stress-relieved first (at a higher temperature than the Tufftride process), and then reground (since it will distort – the amount depending on the stresses encountered during its working life, *eg*

bearing trouble) and finally Tufftrided. After this final process the crank should only be lightly polished with 400-grade and oil. Because of the relatively shallow depth of the process, grinding will remove the compound layer on the Tufftrided 2*l* cranks (and indeed the later nitrided types – post-130 TC).

Whether the process is carried out on a crank requiring a regrind is largely a question of balancing the need for longevity against the estimated working life of the uprated engine as a whole. Certainly, it should be considered on engines of St III, Gp A turbo and above, or where the crank is in need of a 30thou"-plus regrind, or if race con-rod (*eg* 1585) bearings are not available. Fiat/Lancia are not exactly specific as to whether the late 2*l* crank processes are Tufftride or nitride, but this is not significant as Tufftriding is, in fact, a specialized nitriding process. The process will certainly help protect the crank from bearing failure and the contamination of the oil by over-fuelling.

The stress-relieving temperature used is 600°C to 620°C, and the Tufftriding temperature 580°C.

Provided prior stress-relieving is carried out, no really significant journal



8/5: Diagram showing 132 1800 crank radii and journal diameters. (Fiat Auto SpA – copyright reserved)

CRANK GRINDING DATA (source Vandervell)

	ENGINE SIZE (cc) (bore × stroke)		STANDARD		1ST UNDERSIZE (-10thou")		2ND UNDERSIZE (-20thou")		3RD UNDERSIZE (-30thou")		4TH UNDERSIZE (-40thou")	
			(mm)	(in)	(mm)	(in)	(mm)	(in)	(mm)	(in)	(mm)	(in)
1	all 1585	CR	48.224–48.234	1.8986–1.8989	47.970–47.978	1.8886–1.8889	47.716–47.724	1.8786–1.8789	47.462–47.470	1.8686–1.8689	47.208–47.216	1.8586–1.8589
	(84 × 71.5)	M	(as 3)	(as 3)	(as 3)	(as 3)	(as 3)	(as 3)	(as 3)	(as 3)	(as 3)	(as 3)
2	all 1608	CR	48.207–48.227	1.8979–1.8987	47.953–47.973	1.8879–1.8887	47.699–47.719	1.8779–1.8787	47.445–47.465	1.8679–1.8687	47.191–47.211	1.8579–1.8587
	(80 × 80)	M	50.789–50.807	1.9996–2.0003	50.536–50.554	1.9896–1.9903	50.282–50.300	1.9796–1.9803	50.028–50.046	1.9696–1.9703	49.774–49.792	1.9596–1.9603
3	all 1592/1756	CR	50.782–50.802	1.9900–2.0000	50.521–50.546	1.9890–1.9900	50.267–50.292	1.9790–1.9800	50.013–50.038	1.9690–1.9700	49.759–49.784	1.9590–1.9600
	(80 × 79.2) (84 × 79.2)	M	52.984–53.005	2.0860–2.0868	52.730–52.751	2.0760–2.0768	52.476–52.497	2.0660–2.0668	52.222–52.243	2.0560–1.0568	51.968–51.989	2.0460–2.0468
4	all 2*l*	CR	(as 3)		(as 3)		(as 3)		(as 3)		(as 3)	
	(84 × 90)	M	(as 3)		(as 3)		(as 3)		(as 3)		(as 3)	

Notes:
1. Check availability of bearings (VP2/VP19) before grinding.
2. Heat treat crank if going beyond 2nd u/s (especially for St III and above).
3. If crank badly bowed, press before grinding (not cast crank).
4. Stroke correction not vital, consider benefit *versus* cost first. Some crank grinders are not set-up to undertake this. (This process ensures all crank big-ends have the same throw.)
5. 1" = 25.4mm.
6. Measuring: metric micrometer (graduated in 0.01mm) is more accurate than imperial (0.001") type. (0.01mm appr = ⁴⁄₁₀thou"). Always ensure gauge is correctly calibrated (with slip gauges).
7. Grinding to the lower limit for the particular undersize definitely reduces friction and should be the target for an 'ultimate-spec' TC.

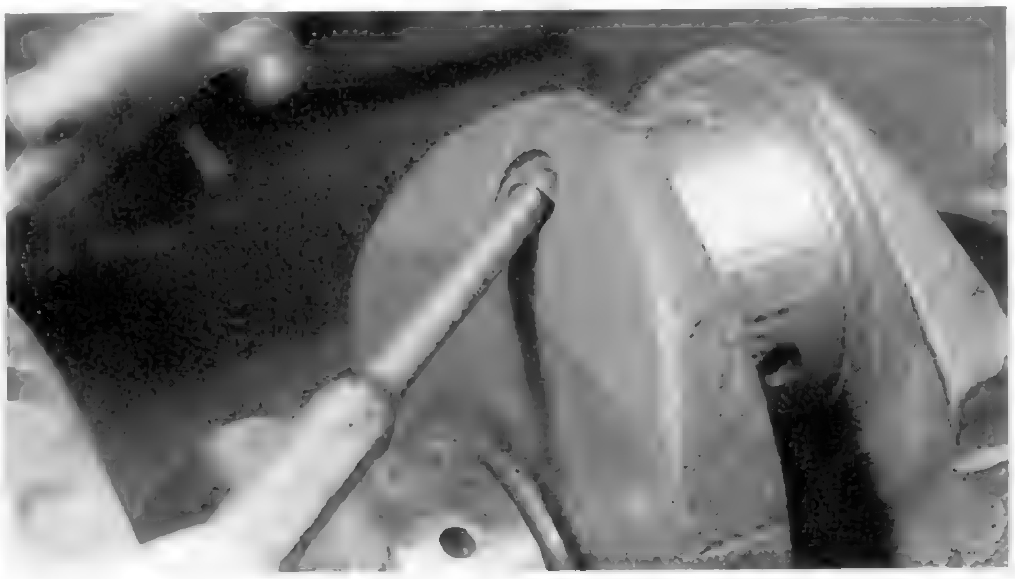
distortion will occur during the Tufftride process.

Tufftriding can lead to a doubling of the fatigue strength of the crank surface (not just bearing journals, but also fillet radii) and is well worth considering for a fully prepared competition TC.

Notes:

1 – Prior to heat treatment, unplug crank galleries and clean, remove crank end bearing (bush can remain in place).
2 – Carry out oilway mods prior to Tufftriding.
3 – Non stress-relieved (steel only) cranks may be straightened by pressing should any distortion occur. This is only permissible if the runout over the length of the crank is less than 0.2mm. Pressing further than this will induce undesirable stresses again. **(8/6 – 8/15)**

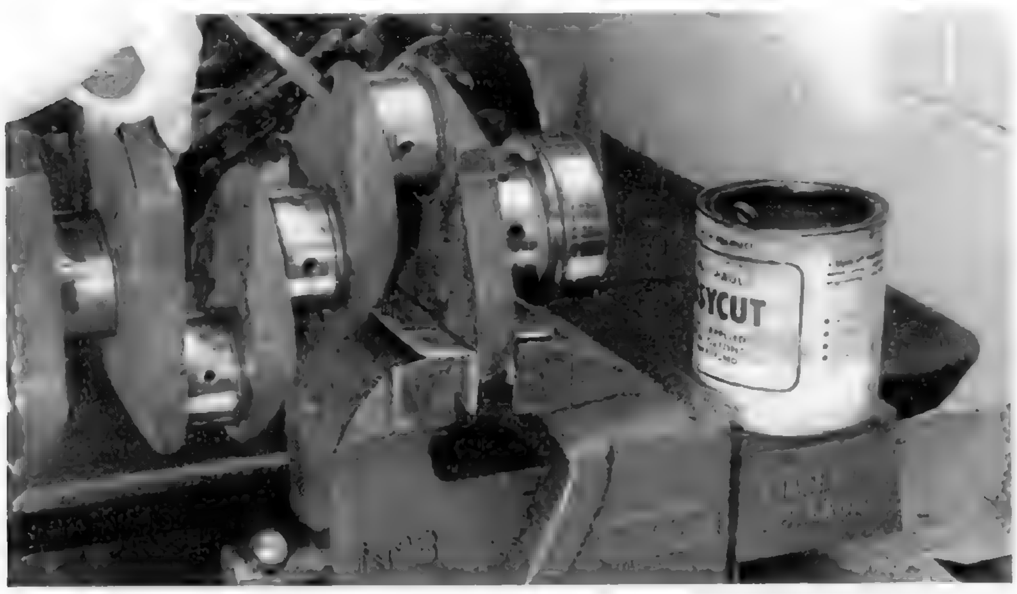
8/6: Crank prep (1): Production 2l 131 crank. Note lack of radius on big-end journal oilway. Crank has been stress-relieved – this should be done if Tufftriding is intended. Grinding and Tufftriding will remove most of rust and scale from stress-relieving. Beadblasting is also effective, but don't use this method of cleaning if oilway threads have already been tapped as grit may become trapped in threads. Wire brush or acid tank is better.



8/7: Crank prep (2): Remove plugs in journal galleries by tapping edge of plug…

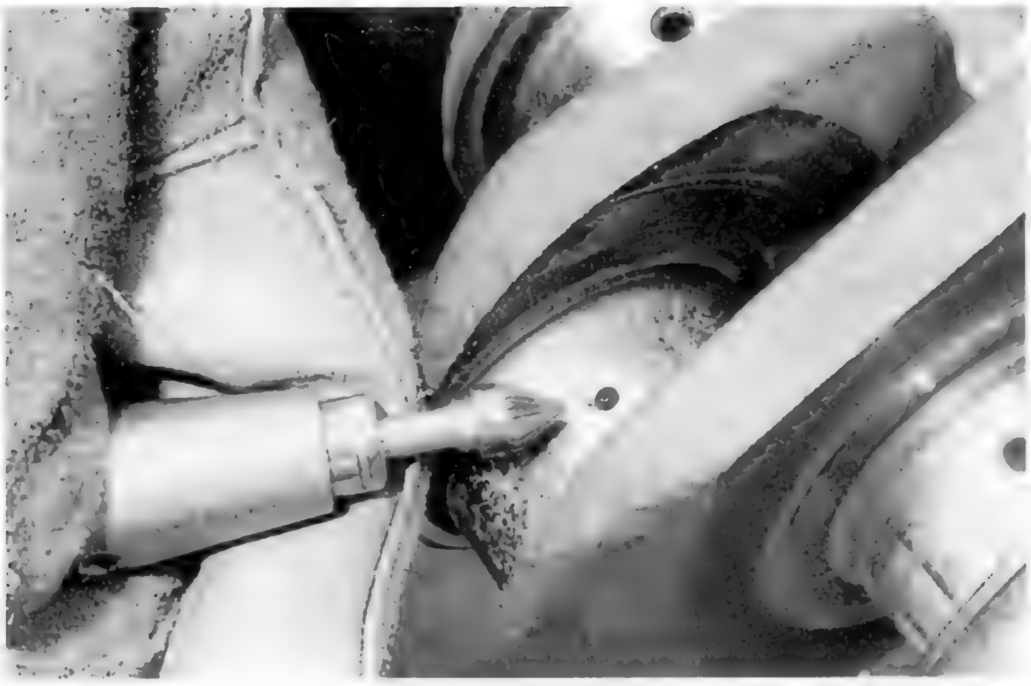
8/8: Crank prep (3): …then tilt plug to remove. All Fiat TC cranks are 'cross-drilled' – a system which improves oil feed to big-end bearings. However, due to centrifuging action, sludge from engine oil collects in end of gallery; plug removal is essential to clean properly. Note rolled radius on main journal.

8/9: Crank prep (4): Loosen sludge with 6mm drill…

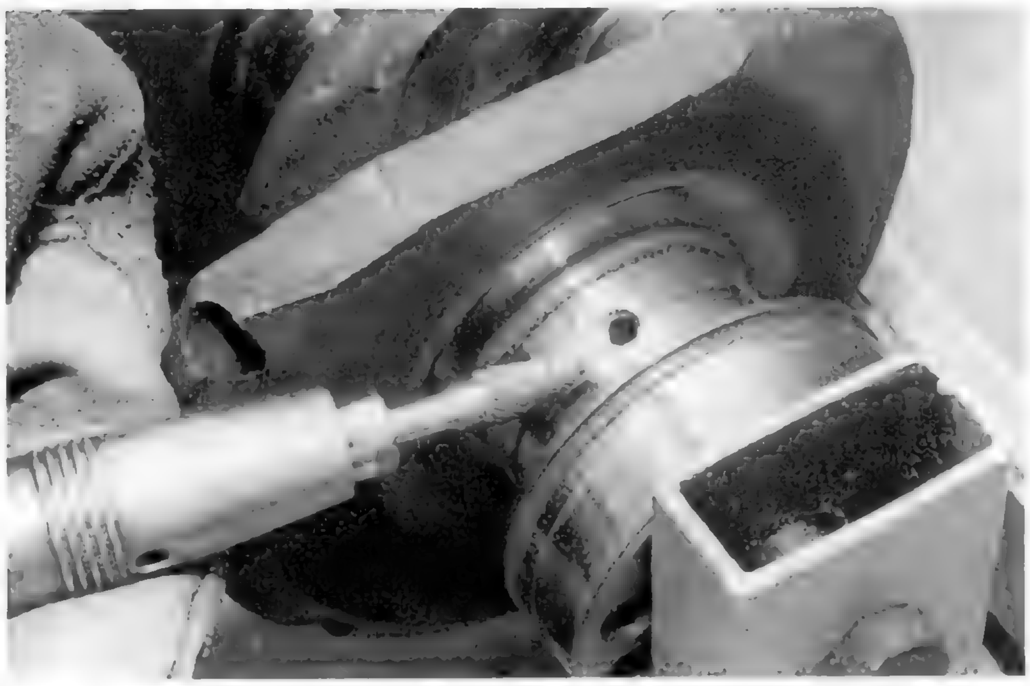


8/10: Crank prep (5): …then tap gallery with $\frac{7}{16}$ UNC tap. Use of cutting lubricant is essential: removing a broken tap from a TC crank is not an easy job!

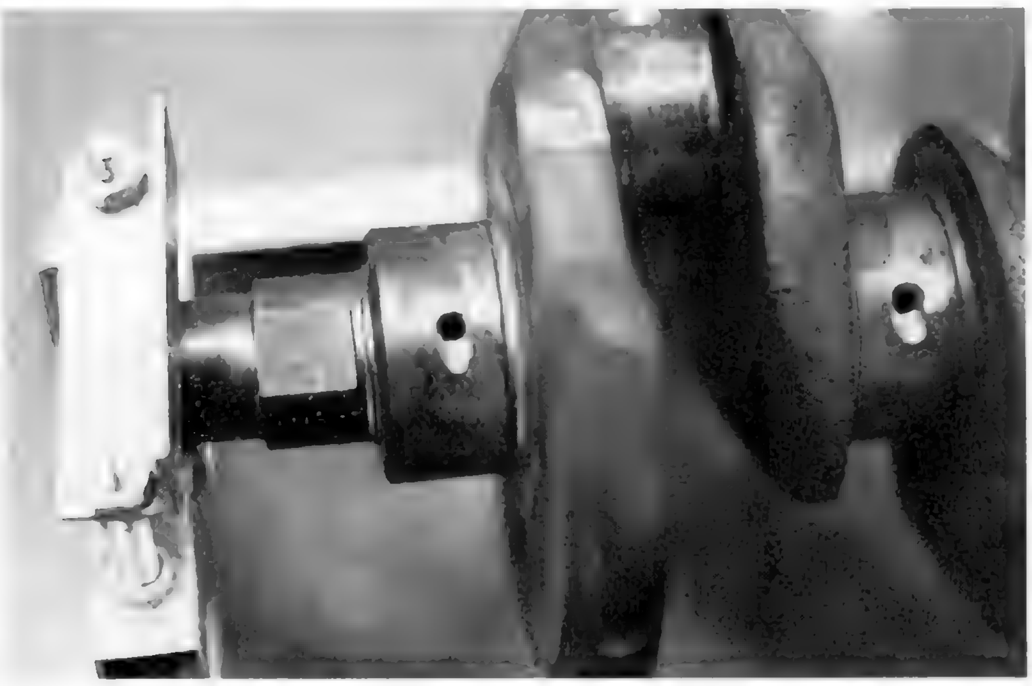
8/11: Crank prep (6): Completed tapping operation. TC cranks have shoulder about 1cm inside drilling – do not attempt to tap further; ½" long socket grub screw and Loctite Hydraulic Lock seals thread.



8/12: Crank prep (7): Use of die grinding tool and countersink to improve exit radius on big-ends. Do this before grinding or lapping crank! Gloves protect hands from dangerous steel splinters. Countersink shown has 60deg included angle – 90deg is better. Lack of radius will dramatically shorten journal life on a crank that has not been heat-treated or where the hardened layer has been ground off.



8/13: Crank prep (8): Pear-shaped cutter used to keyhole main journal oil holes to improve pick-up of oil from bearing. Note direction.



8/14: Crank prep (9): Showing basic shape of main journal keyholes on stress-relieved crank. Crank rotation in engine is clockwise on all TCs viewed from crank nose (front pulley end).

8/15: Crank prep (10): Finally, dress holes with fine grinding point to reduce stresses caused by sharp edges. Crank is now ready for grind or lap.

Crank bearings

Some TC cranks have 'odd' factory undersize journals, for which bearings are not available.

The size (*eg* +0.05) is usually stamped on the adjacent crank web, or marked on the old bearing shell. Such a crank will have to be reground before refitting (unless the old bearings have been marked and are to be re-used!). Always 'mike-up' the old crank to check (*ie* use a micrometer!).

Bearings by Vandervell, Glacier, Clevite and Federal Mogul are made to some of the highest conformance standards in the industry and, provided the housings and cranks are of the correct size, will fit perfectly every time. Bearings are designed to be diametrically slightly larger than their housing so that when crushed into position during torque-up they are unable to rotate.

Two main types are available for the TC, as follows:

Vandervell VP2 – steel-backed, leaded bronze overlay infused with indium, added to resist corrosion. These bearings have a design loading of 7000lbf/in^2, but in Vandervell lab tests have withstood over 15,000lbf/in^2.

Aluminium/tin (Vandervell VP19) – steel-backed, aluminium/20% tin overlay. Design load is 4500lbf/in^2.

BEARING SELECTION

1585	M	VP2 mains up to 20thou" oversize available from Fiat/Lancia (part no is from 130 TC or Volumex). VP19 also available
	B	only VP19 type rod bearings available
1756 (124 Sport)	M	as 1585
	B	VP19 only now available (asymmetrical con-rod spray holes)
1756 (Fiat 132) and all 1995 except late 8v (reversed-port) and 16v models	M	as 1585
	B	VP2 available (as 130 TC) up to 20thou" o/s from Fiat. No longer available direct from Vandervell. VP19 also available (symmetrical oil spray holes)

Vandervell used to make steel-backed lead/bronze thrust washers (VP10) but these are no longer available; VP19 types must now be used. Some Fiat/Lancia models used VP2 (eg 130 TC, Volumex). Most other early models used FM VP19 equivalent. No data is available at time of writing on bearing types for late turbo models.

Bearing selection

The supply of bearings for the 1608, although race main (VP2) types were formerly listed by Vandervell, is now extremely limited. It must be assumed that only limited stocks of VP19 remain.

All late (reversed-port) 8v and 16v Fiat/Lancia engines (*ie* and turbo except 1600 Delta Turbo *ie*) require the use of OE Fiat/Lancia bearings due to the redesigned locating lug.

Thrust washers

All the 84mm bore TCs share the same design. Only VP19 is now available. Standard, +0.0025" and +0.005" are supplied. 1608 types (125, 124 BC) use a different size (same as X1/9).

Crank cleaning

Removal of the oilway plugs is the most effective way to clean the TC cranks (which are all cross-drilled). If this is not carried out there is a risk of foreign material (from grinding or balancing) being trapped in the sludge in the end of the gallery. Preliminary cleaning prior to preparation can be with any commercial solvent (Comma Hyperclean, Jizer, etc). When all preparation is complete, the crank should be washed again, preferably rinsed with hot water (under pressure to clean galleries thoroughly) and detergent, followed by drying with an air line and final clean with a trichloroethylene-based solvent, *eg* ICI Triklone. Last of all, when the oilway plugs have been installed, coat the crank with aerosol oil, *eg* WD40.

Con-rod reconditioning

It is not uncommon to see a twisted TC rod, but if they are checked on a special jig or surface plate and found to be out-of-true, provided the degree of twist is not excessive they may be safely straightened.

Checking this parallel relationship between the big and small-end bore axes requires the use of a special jig; the factory allowance is ±3thou" measured 4.92" from the rod shank. These two measurements should not be a cause for concern unless there is damage evidence on the bearings that something is seriously amiss.

Leave the small-end bushes in place if they are in a smooth, well-polished condition; the fit between the piston pin and the bush should be 0.0004"–0.0006". Light honing with 400-grade is worth doing to improve oil retention.

More important is the size of the big-end bore:

Engine	Big-end bore diameter (mm)
1585cc	51.330–51.346
1608cc	51.330–51.346
1592/1756cc, 2*l*	53.897–53.913

Resizing con-rods is perfectly acceptable and is carried out by grinding the big-end cap parting faces to reduce the diameter, then honing the bore to the new size with the cap bolted in place. Because of the accuracy required, this work, as with crank grinding, must be entrusted to a reputable expert. GCT have run 'in house' engines (*eg* the author's 124 Abarth Spider) with the rods 2thou" out of spec; the penalty is certainly lower oil pressure, especially at tickover (the author's car had 12lbf/in²). (If much metal is removed during resizing, the set-up height of the pistons will change – check before building.)

If the small-ends need to be replaced (new bushes are available from Vandervell and Fiat) the old ones must be pressed out with a drift of suitable size, the new ones pressed in and then reamed (or preferably honed) to the correct size. This is a tricky operation and should not be undertaken unless there is a compelling need. If the bushes are accidentally reamed oversize they will rattle abominably! GCT have only seen bush failure on one engine – a very highly stressed Gp A 8v Integrale, when the wear was merely due to the exceptional loading involved. The finished diameter of the small-end bush should be 21.996–21.999mm (production pins vary from 21.991–21.994 Class 1 to 21.994–21.997).

CON-ROD MODIFICATIONS

Lightening

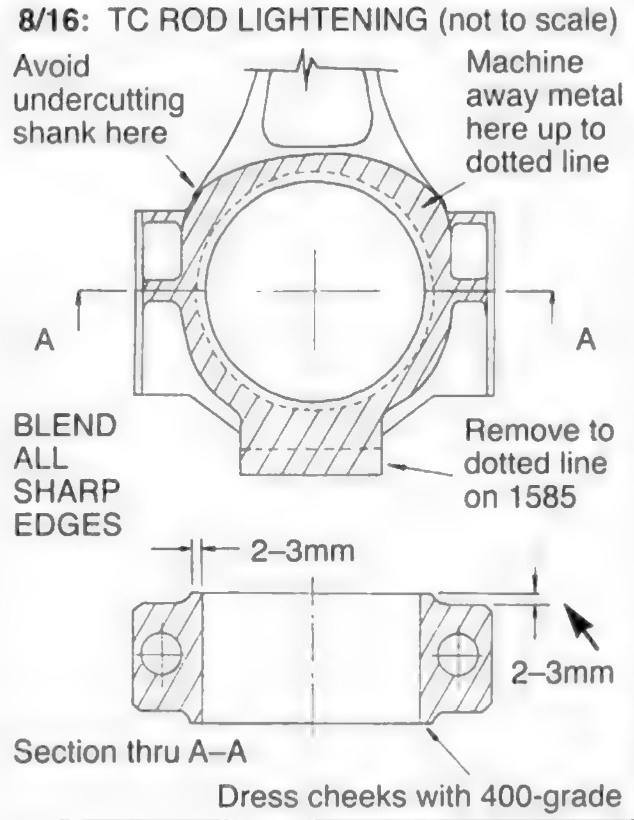
The 1585 rods have a heavy balancing pad on the big-end cap which can be removed by milling until the cap thickness is around 10–12mm. Unless the rods have been balanced previously, leave enough metal on the end cap for this. On all the TC rods, the width of the small-end can be reduced by 4mm (this *must* be done by grinding or the small-end bush will be disturbed) and the forging flash along the shank removed with a die-grinding burr. Lightening of the big-ends can be carried out by mounting the rod in a lathe and machining as shown. Put an old bearing in place to protect the bores from damage by the lathe chuck jaws. This can also be carried out on a mill using a radiused slot drill and dividing head. (8/16, 8/17)

The inertia of the rod contributes to torque loss at the flywheel, so lightening should release more usable power (although GCT have no figures to quantify this) if the budget permits.

Shot-peening (8/18)

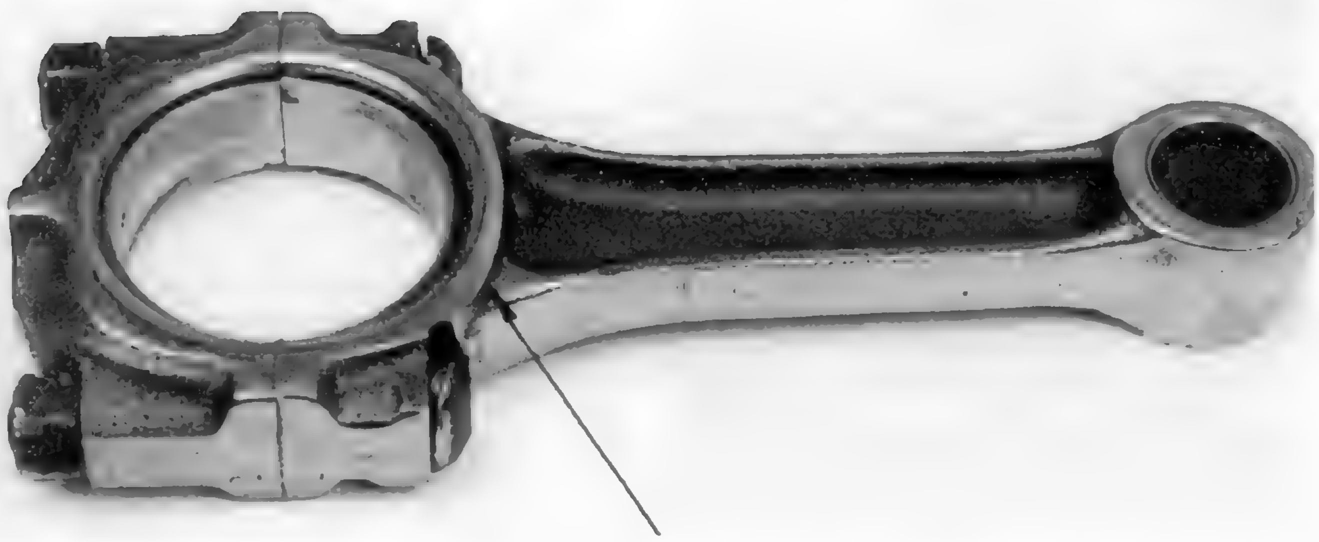
In order to maximize the benefit from peening with steel shot, the rods should be part-polished first down to a 80–120 grit finish. The key areas for treatment are around the sides of the small-end, the shank (especially the sides), the root area around the rod shank and the base of the big-end cap. The rod can, of course, be shot-peened in other, less crucial areas – to no benefit! Shot-peening (with proper preparation) gives at least a 20% increase

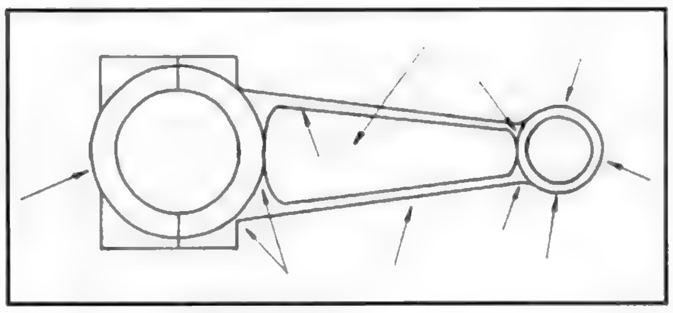
8/16: TC ROD LIGHTENING (not to scale)



8/17: A lightened 1600 rod.

8/18: Use of rubber-lined beadblasting cabinet and special pressure pot allows shot-peening operation on rods and 20%-plus increase in fatigue life. Protect small and big-end housings from shot damage. Beadblasting with Guyson's Honite (spherical ceramic beads) cleans and prepares surface of alloy components prior to build-up. Especially good for cleaning carbon deposits from heads – but first make sure they are dry and oil-free. A good machine with recycle filtration and water trap on airline circuit will extend life of media.





8/19: Incorrect polishing technique: rod must be polished along shank, not across. Key stress area (arrowed) has not been polished at all, nor stress raiser around bolt head. Cheek of rod in contact with crank must be left with 'ground' finish for oil retention. Damage to polished area is evident. Note condition of nuts – engine had supposedly been 'rebuilt'. Beware of 'polished' rods, which look good – but have been 'done wrong'!

Drawing shows con-rod key stress areas (arrowed) for partial polishing and shot-peening. Ensure all machined faces are protected from shot (small-end bush, big-end bore and cheek of rod). It is best to shot-peen with rod split to gain access to area around bolt head.

in strength and consequently fatigue life because the impact of the shot compresses the outer skin of the rod. The compressive stress induced resists the tensile loading on the rod during its reciprocating motion.

The stock forged finish can be improved by full polishing, though any increase in fatigue life is questionable, especially in view of the cost involved. Certainly, once a rod is polished (220 grit is appropriate), care must be taken not to scratch the surface or crack propagation may result.

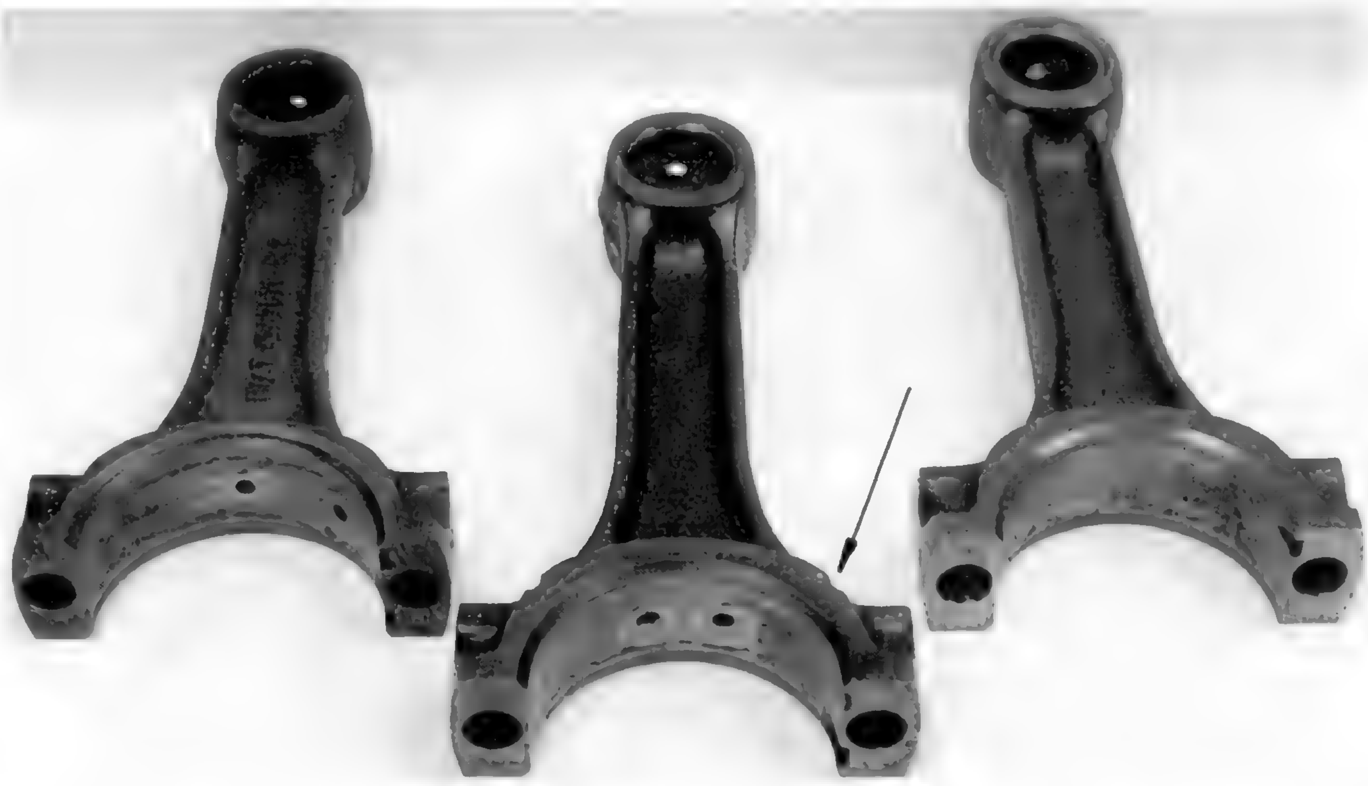
GCT have employed standard, unmodified rods on all the TCs at very high levels of tune and rpm (including a Gp A Integrale with 27lbf/in^2 boost) with no problems, indicating that their factor of safety must be *very* high. There is absolutely no requirement to resort to specially made versions up to National level motorsport. (8/19, 8/20)

Balancing

Cranks, front pulley and flywheels are balanced independently at the factory, then, with the assembly married together, final adjustments are made. The standard of factory balance is exceptionally good.

Imbalance is measured in terms of gm.cm of torque effect produced by the imbalance force during rotation and is independent of crank rotational speed, though of course the stress created in the rotating assembly (crank journals, crank bolts, flywheel and, to a lesser extent, bearings) is cyclical and has a major influence on the fatigue life of the components in that the higher the crank speed the more cycles of stress per second are exerted on the running gear.

From this point of view it should be appreciated that an assembly which is run out-of-balance may survive a considerable length of time at modest rpm, but then suddenly fail at a higher level, causing the flywheel bolts to shear (with possibly



8/20: Con-rods. Left to right: 124 Sport 1800 with offset oilways, 131/132 2l (132 1800 also had symetrically placed oil spray holes), and 8v Integrale (switched to bolt-only fixing). This eliminated stress raiser caused by machining root of rod shank for rod bolt (arrowed). Note oil hole in small-end; this must be redrilled when new bush is fitted. All TC rods are forged steel and exceptionally strong and will accept a certain amount of lightening of small and big-end without reduction of fatigue life. Rods do go 'oval' in service, but can be resized by grinding mating face of caps to reduce diameter, then honing back to shape. Very involved 'works' rod prep on Abarth 124 and 131 involved lightening, polishing, shot-peening and Tufftriding, followed by new bushes and honing. Increase in fatigue life must have been around 100%.

8/21: Broken flywheel bolts caused by failure to tighten clutch bolts. Although bolts will withstand very high shear in torsion, their layout will not withstand fatigue load induced along axis of bolt by flywheel imbalance. Note balance holes on flywheel rim. Clutch dowels are not yet in place. Flywheel friction face has been reground; this is preferred to refacing in lathe – especially if there are 'hard spots' caused by clutch slip.

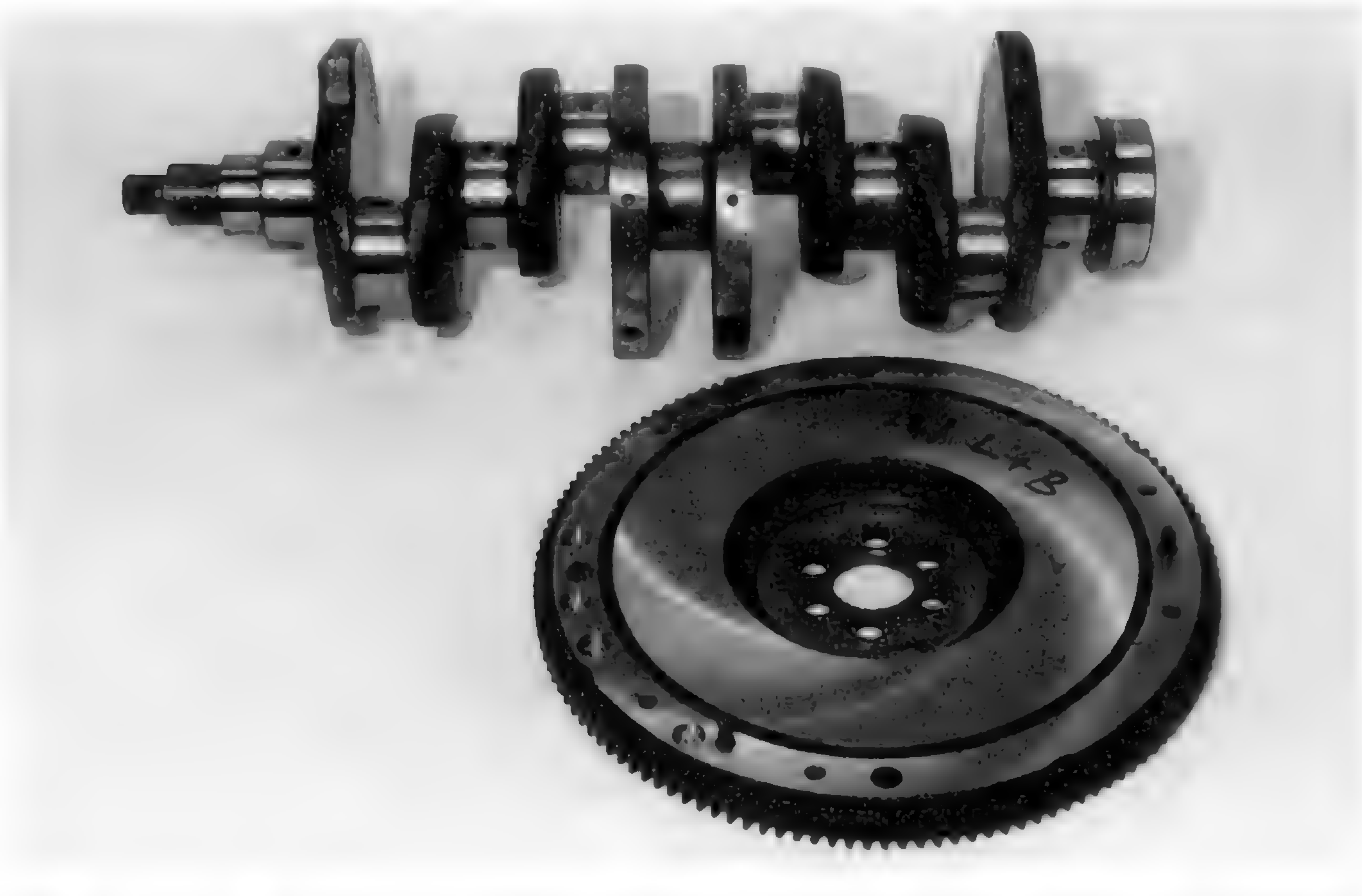
catastrophic results, *ie* death or serious injury). (8/21)

A crank which is slightly bowed can be ground true, but the resulting deflection on the driving flange, albeit small, when amplified by the radius of the flywheel may well cause bolt failure. Balancing of the flywheel/crank assembly is *vital*.

Similarly, if the flywheel is lightened, modified, or to a lesser extent refaced, the same procedure must be followed. A good balancer *(see Recommended Suppliers)* can detect the imbalance caused *even* by the *density* changes in the material of a cast-iron flywheel uncovered by refacing. Static balancing between knife-edges is not sufficient; only dynamic balancing will reveal the position of the imbalance force somewhere along the length of the crank/flywheel assembly. This work is best entrusted to a *proven expert*; the consequences of a sheared flywheel can be extremely grave. Never 'take a chance' with balancing, and for this reason, never run an engine at high speed without a protective bellhousing fitted. (8/22)

The balance of the standard factory rods (as a matched set from a donor engine) is reasonable. Fiat in fact quote a maximum deviation between rods of 3gm, although *(see later)* GCT have seen them worse!. If the rods are not to be polished or lightened and they match this 3gm criterion there is no virtue in balancing them.

Many engine builders claim to balance rods to ±0.5gm; this is perhaps important on a 'V' engine where a balance factor has to be established so that weights equivalent to the load exerted by the rod (in service) on the crank can be attached to it for balancing purposes, but on the in-line 4cyl TC, GCT have run rods as much as 8gm out with no detrimental effect whatsoever. Whilst this seems excessive, it must be remembered that it was purely for experimental purposes! A variation of ±2gm is a perfectly reasonable figure to aim for, bearing in mind that the random weight of oil clinging to a rod is probably greater than this.



8/22: Crank prep. Reground 124 1800 crank and flywheel balanced and ready to fit. Note new balance holes (lower left) on flywheel and metal removed from crank centre webs. Seal faces on crank nose and rear flange have been dressed-up with 400-grade carborundum paper on grinder – but this can be done by hand.

Fiat recommend in their workshop manuals that to balance the rods 'end-over-end', ⅓ of the excess weight should be removed from the small-end and ⅔ from the big-end. This method works surprisingly well, but the optimum method is to suspend one end of the rod and weigh the other. Before the invention of digital scales, this process was quite difficult because the deflection of conventional scales caused one end of the rod to drop, thus changing the effective torque applied to the scales by the end being weighed, owing to the slight shift in the position of the centre of gravity of the rod. (8/23, 8/24)

The procedure should be as follows:

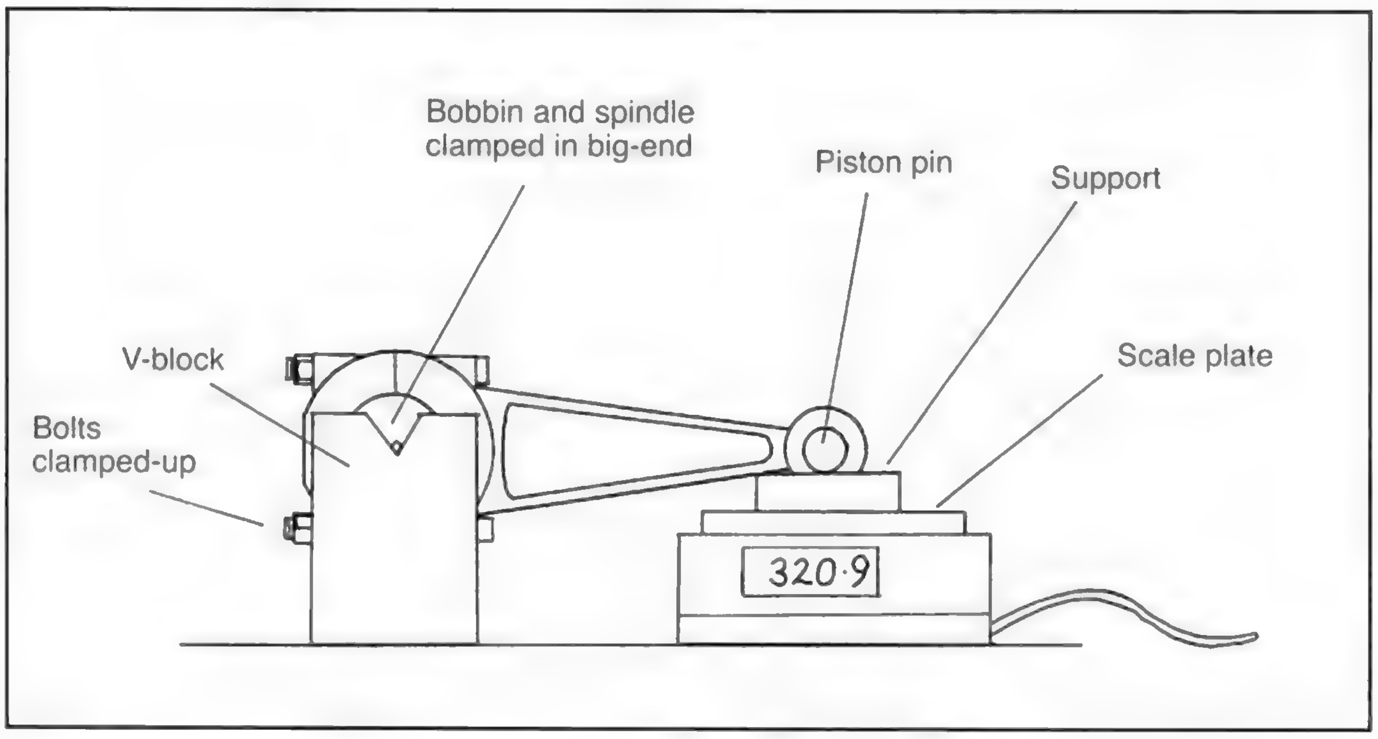
1 Weigh the rods overall (with nuts and bolts)
2 Weigh all the small-ends
3 Weigh all the big-ends

The process involves removal of metal from the small or big-end as required until all the end weights are the same weight as the lightest. When removing metal, if, for example, one big-end is 9gm lighter than the others, do not remove 9gm, but only remove ⅔ of this amount, *ie* 6gm, since the centre of gravity of the rod shifts as metal is removed and removal of the whole 9gm mass 'in one hit' will throw out the small-end significantly. Be careful not to reduce the thickness of the small end excessively.

Systematically remove and reweigh, and eventually, when the end-over-end weights are the same, the overall weights will also finish equal and the C of G of all four rods will be in the same place. This contributes to the smooth running of the engine and reduces crank stress.

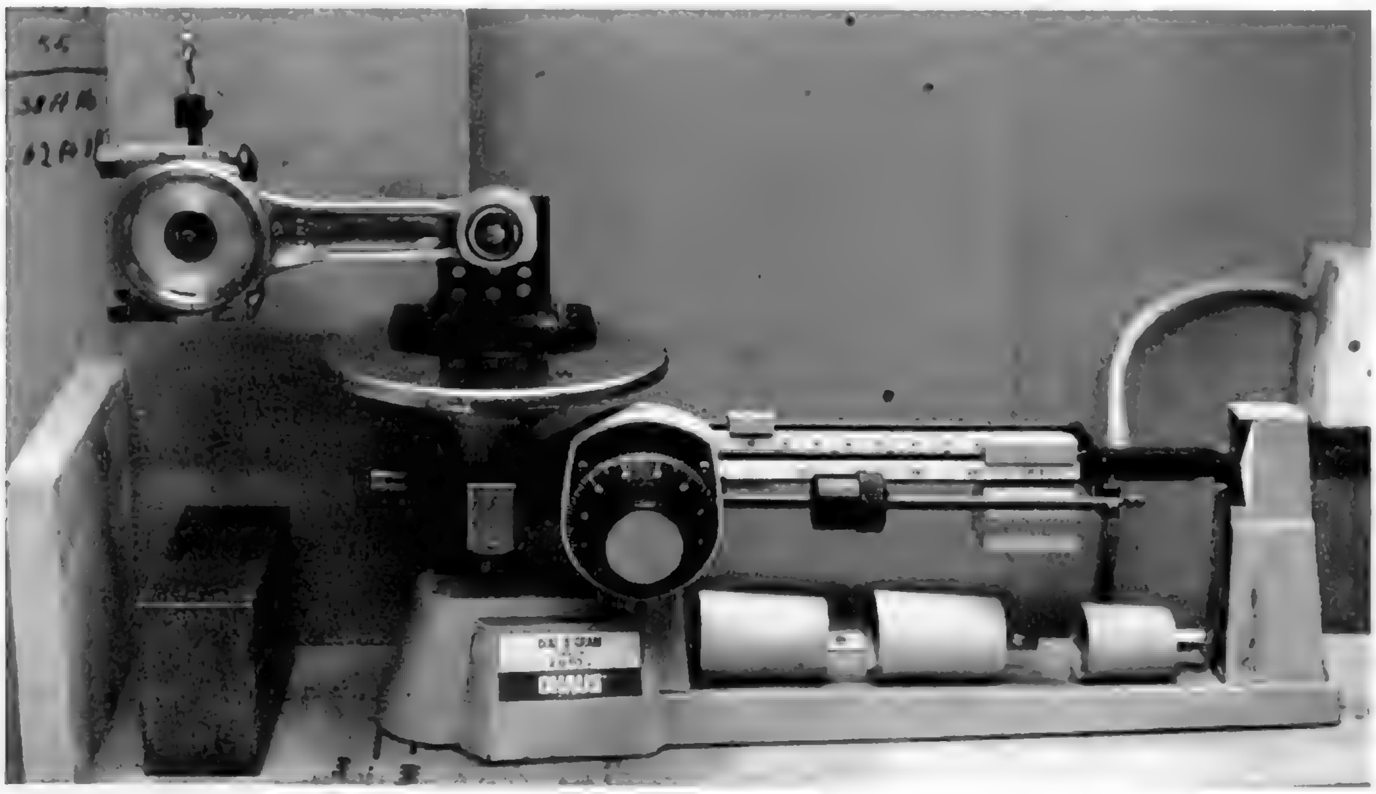
FLYWHEEL RECONDITIONING

The most obvious reconditioning task is to clean-up the friction face. This is best done by grinding (a special machine is used by most shops). The step height of the friction face must be maintained and it is usual to finish the outer periphery of the flywheel with a lathe. The step should be 20thou" ±2thou" and a step of the correct height ensures that the clutch cover clamp load will be correct. It is important to maintain the parallel relationship between the flange face (where it bolts to the crank) and the friction face and the overall trueness of the friction face itself. Fiat quote a maximum runout of 4thou", but it is perfectly possible by machining to reduce



8/23: CON-ROD BALANCING WITH DIGITAL SCALES
Negligible deflection of scale plate with digital scales allows big and small-ends to be weighed simply with system shown.

8/25: Lightened cast flywheel. This is the maximum recommended amount that should be removed from TC flywheels. Note generous machining radii. Cast iron flywheel must be kept about 1cm–12mm thick for strength. A lightened flywheel gives a significantly quicker throttle response.



8/24: Simple balancing rig uses needle roller bearings in big and small end. Provided bearings are clean and lightly oiled, repeatability is very good.

this to less than 1thou" with an accurate lathe or grinding machine. A flywheel badly machined will mean more work for the balancer and extra load on the central section of the flywheel every time the clutch is actuated. (Do not use refacing as a 'lightening' operation, as the clutch plate may foul the flywheel bolts if too much metal is removed.)

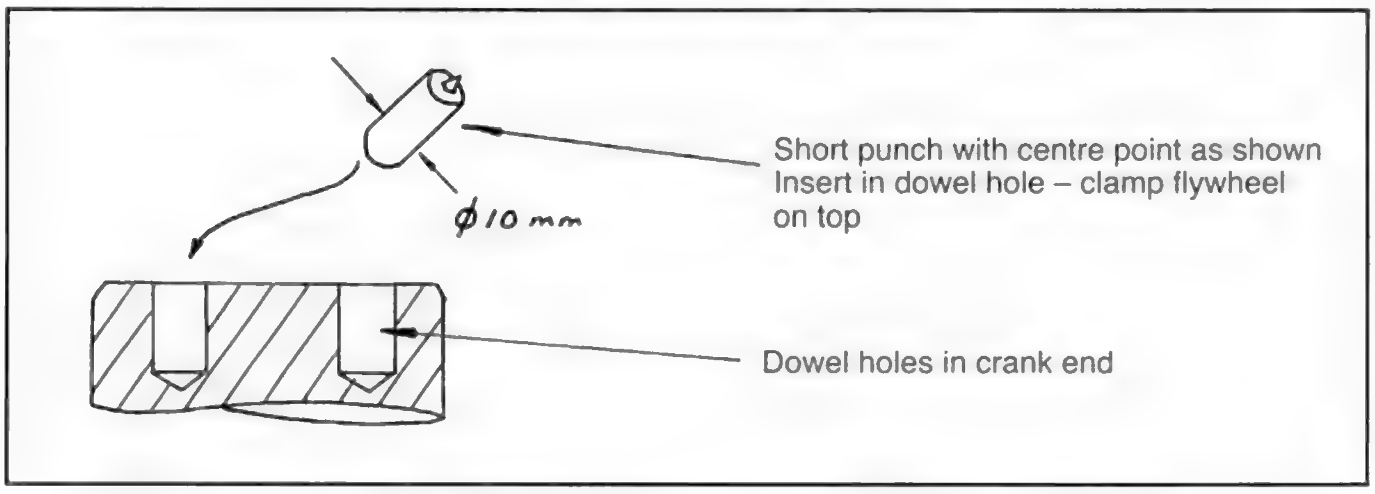
Check the condition of the ring gear. If badly worn it can be replaced; grind through the hardened outer skin of the gear and drill through it with a 6mm drill. A sharp cold chisel will then split it and it can be removed. Alternatively, it can be pressed off with a hydraulic or flypress. When fitting the new gear it should be preheated to 80°C in an oven or with a torch and it should then slip on quite easily. Remember to fit with the bevel on the ID towards the flywheel.

Flywheel modification

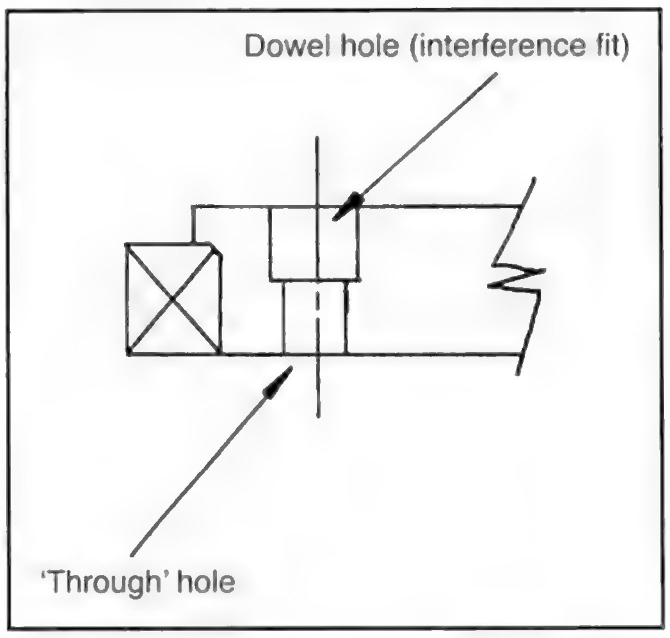
The standard flywheel can be substantially lightened (8/25) and this improves throttle response and minimizes power loss. A heavy flywheel absorbs substantial power because of its inertia. GCT have not encountered any low-speed smooth-running problems with light flywheels, indeed the Guy Croft Formula 2000 hydroplane used only a thin steel disc (from an automatic car) with ring gear and ticked over quite happily at 900rpm (with a full-race 1800 engine). Throttle response was instantaneous...

Cast flywheels should not be used on engines above 7200rpm. Above this speed, steel flywheels must be used because cast types are liable to crack and break loose. It is critical that cast flywheels are not lightened excessively (see photo) or they will be too weak. Above the speed shown (which equates to St II engines) also include dowelling the flywheel to the crank as part of the operation. All the TC cranks are ready-drilled for 10mm × 20mm dowels, which can be bought in stainless steel from Unbrako or other suppliers, or made from old head bolts.

The easiest way to mark the flywheel for drilling is either to 'blue' the crank with engineers' blue and clamp the flywheel to it (the centre of the marks can be found with dividers), or to employ a shortened 10mm punch as shown (8/26). The method used at GCT is to bolt the flange from an old crank (cut off and



8/26: Marking flywheel for dowels



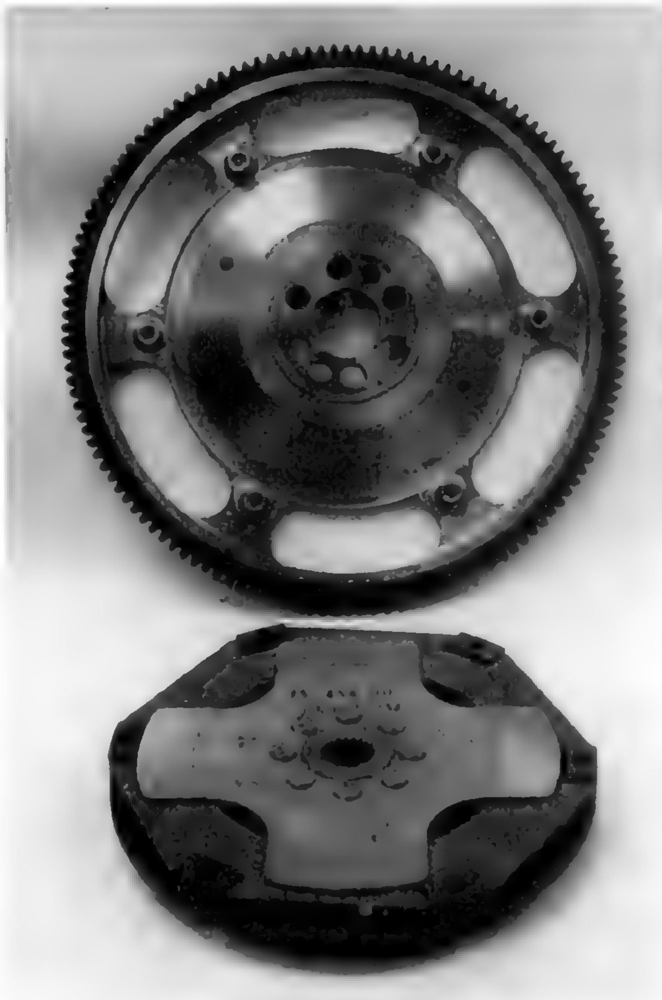
8/27: Redrilling dowel holes in flywheel

ground true) to the flywheel – this acts as an accurate centre for drilling the 2 × 10mm holes required. It may be necessary to dress the dowel holes in the crank (especially after Tufftriding) to ensure the dowels fit easily. Do not hammer them in – they will never come out again! (GCT bolt the flywheel to the crank and check-ream the holes with a 10mm reamer.) (8/27, 8/28)

On all the TCs the flywheel is centred on the crank axis by the bearing in the end of the crank (RWD – to support gearbox shaft) and by the bush on FWD models. This must not be omitted or the flywheel will not 'centre' properly. The bolts are a loose fit in the flywheel, and dowelling ensures a tight fit between the crank/flywheel, thus reducing the load on the bolts. Under torsional loading, the flywheel does shift slightly, even if the bolts are accurately torqued. Dowelling prevents this and eliminates any risk of fretting around the bolt holes.

Conversion of TC flywheels to accept other clutches (*eg* RS2000 Ford 1" × 23 to mate up to the Sierra five-speed box) can be straightforward provided the diameter of the flywheel is large enough to accept the new clutch plate and cover. The RS2000 Gp N clutch (uprated organic), for example, requires the flywheel face to be machined flat (step removed) and the outer edge of the flywheel drilled and tapped for dowels and clutch cover bolts at the appropriate pitch circle diameter.

In order to calculate the new dowel positions (the bolt holes can be marked through the cover once the dowels are fitted) some simple geometry is required (8/33). In this case, dowels from the Ford would be used. Remember to drill a 'through hole' so the dowel can be removed easily if required, then counterbore the finished hole for the dowel to an appropriate interference size to hold it in place (1thou" undersize).



8/28: Ultra-light En8 steel flywheel (reverse view) and Sachs 7¼" plate. Flywheel is machined on lathe from profile, flame-cut from solid plate. Key stress area around centre of flywheel is minimum 8mm thick, outer edge (holding ring gear) only 4mm. Lightening slots are machined with mill.

This particular flywheel has been slot-lightened by CNC machining, resulting in a product with near-perfect concentricity and balance. Lightest available ring gear is 130 TC (2l) type, secured by combination of shrink-fit and 4mm dia grub screws at 120° intervals. Note additional two holes in centre for dowels and drain/cooling holes approx 2" from flywheel centre. To avoid necessity for nuts on clutch retaining bolts, six threaded bosses have been machined on rear of flywheel. Single-plate clutch has a very high torque rating, but use is limited to strictly 'in-and-out'. Unlike conventional organic type, sintered facings on 7¼" disc will not tolerate slip. Cerametallic or sintered facings have a coefficient of friction considerably higher than organic material. Race flywheels can be Tufftrided, but machining is more involved. This process reduces frictional coefficient of flywheel face and can cause distortion due to high temp involved. Technique is: machine to within 4thou" of finished size, stress-relieve, finish machine, Tufftride and finally grind 12–15μm from friction face (maintaining step height) to remove compound layer. Only purpose of Tufftriding flywheel is to improve fatigue strength (by up to 100% on high chromium content steels).

Carry out this flywheel machining operation before lightening to ensure that sufficient thickness remains in the flywheel for the new cover bolts. As a general rule, on cast iron the thread penetration should be 1.5 × thread diameter, *ie* for the 8mm bolt the flywheel must be 12mm thick in the thread area. Steel flywheels can tolerate 8mm thickness in the flange/bolt area and as thin as 4mm in the low-stress area between the friction face and clutch bolt holes.

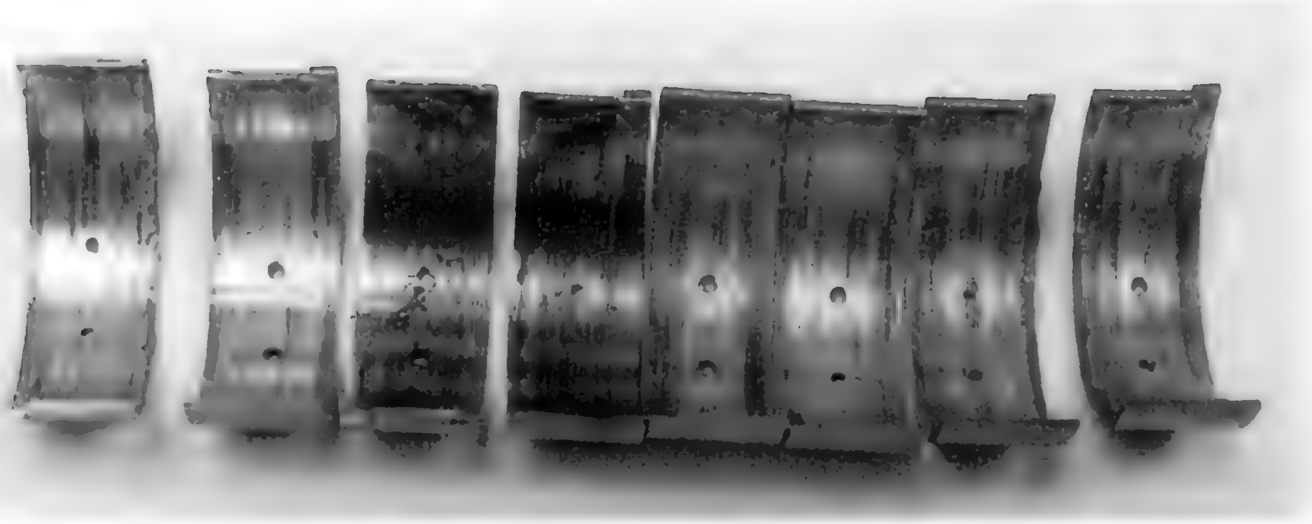
8/29: If con-rod bolts are press fit in rods, vice and old gudgeon pin provide easy way to fit them. Failure to press fit correctly may upset torque setting. Bolts shown are 130 TC.

In summary, the procedure for flywheel mods should be:

1 Grind friction face
2 Machine step
3 Redrill/tap as required
4 Lighten
5 Balance as described earlier

Note: Clutch balancing – TC bolt-on clutches by Sachs, Valeo, Fiat, AP Borg and Beck and other reputable companies do not need to be balanced with the clutch/flywheel assembly provided the same rpm limit as for cast pistons are adhered to (above these limits, balance the whole assembly), but modified set-ups (*eg* Ford-Fiat or 7¼" race conversions) *must*. As a general rule, if in doubt, balance!

8/30: Use of 130 TC crank with 131 RWD gearbox led to demise of thrust washers and bearing/crank damage on this GC 2l engine. Beware! Recess in end of 130 crank is not deep enough for RWD box input shaft. Answer? Shorten input shaft with angle grinder or put spacers between block and bellhousing. Crank was properly fitted with sealed end bearing (and sleeve over bearing to fit flywheel bore) but input shaft bottomed-out. Constant contact of shaft-end with crank led to chronic overheating of shaft, bearings, seal and crank.



8/31: Damaged aluminium-tin bearings show how surface starts to break up under contact with crank. Chronic overheating oil was cause. Damage of this type will cause rods to go severely (8thou"-plus) out-of-round.

Fasteners
Always use new flywheel bolts when rebuilding a TC. This is an inexpensive precaution against bolt failure. Only original equipment Fiat/Lancia 12.9-grade bolts should be used, together with the original spring steel plate under the bolts. Clutch bolts should be 8.8-grade minimum, secured with new spring washers.

Always replace con-rod nuts and bolts with new original equipment items, unless (as with other fasteners) the recent history of the engine is known to have been favourable and the items concerned fall within the GCT life schedules (see *Appendix A*). Never re-use con-rod fasteners if the engine has suffered bearing trouble. **(8/29, 8/31)**

Crank end bushes and bearings (8/30)
Types vary. Fiat 131 types (RWD) use sealed ball-type ID 15mm, 12mm thick, 35mm OD. All FWD cars are bushed. Conversion of, say, a 130 TC to RWD requires a smaller bearing with a sleeve to fit the flywheel: the standard crank has a stepped bush. Make sure the input shaft does not foul the end of the crank *(see photo)*. The easiest way to remove is to pump the end of the crank with grease and force the bush out hydraulically with a close-fitting drift. Hit the drift with a heavy hammer and grease pressure will force it out. Stubborn bushes can be threaded and an appropriate bolt screwed in to release. Bearings can be removed in the same way provided the seal is intact, otherwise a puller must be used.

8/32: Wedged 2l crank with rods prepared to 730gm as described.

WEIGHTS
Standard 1600 (1585) rod: end-over-end analysis
(Weights shown include the weight of the balance rig and the con-rod nuts/bolts, but not big-end bearings.)
1st weigh

Rod	Big-end (gm)	Small-end (gm)	Overall† (gm)
1	995*	509.5	1013*
2	998.5	511.5	1017
3	1003.5	507.5*	1019.5
4	998	510	1016.5

† (deduct 300gm for true weight)
* lightest

1st removal of metal (from the small-ends only) equalized the small-ends, but affected the big-ends (due to the centre of gravity shifting towards the big-end).
2nd weigh

Rod	Big-end	Small-end
1	996	507
2	999	507
3	1003.5	507
4	998.5	507

2nd removal to equalize the big-ends slightly upset the small-ends, as shown.
3rd weigh

Rod	Big-end	Small-end
1	996	507
2	996	507.5
3	998	508.5
4	996	508

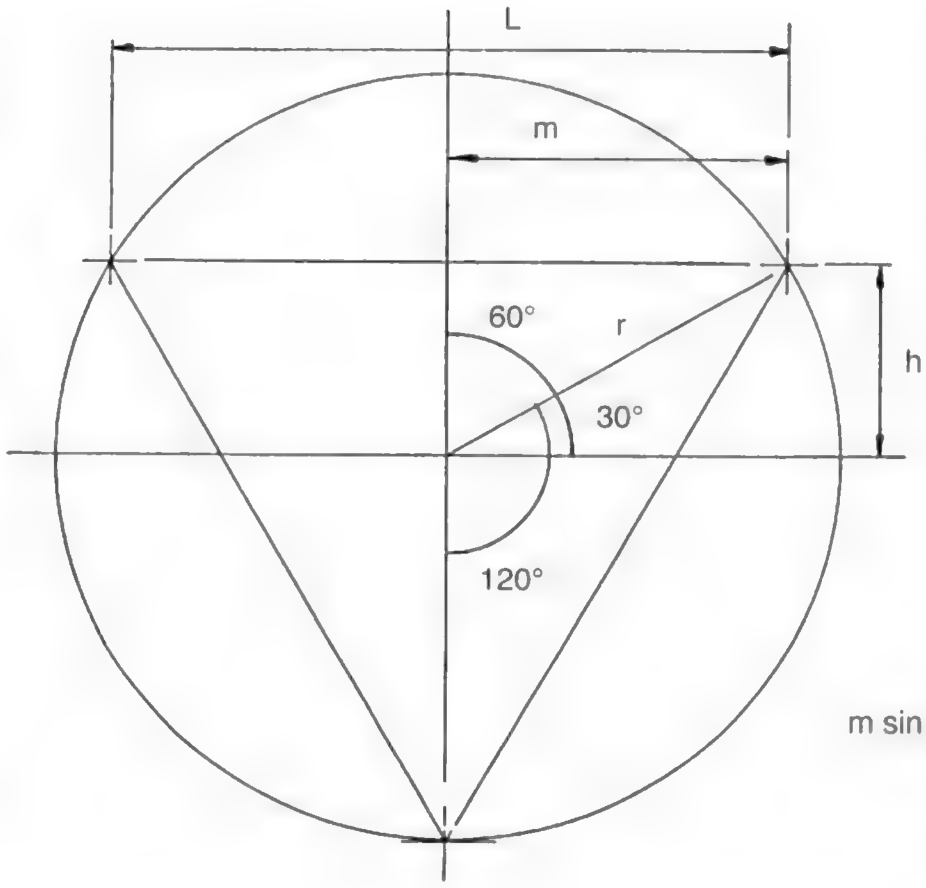
Note that at this point the small-end of the No 3 rod has gone 1.5gm over, while the big-end is still 2gm over the others. Further removal of metal from No 3 big-end would 'throw out' the small-end even more. It is questionable whether it would be worth pursuing such a minor disparity in view of the time involved. Final deadweights showed a mean of 1014gm ± 2gm.

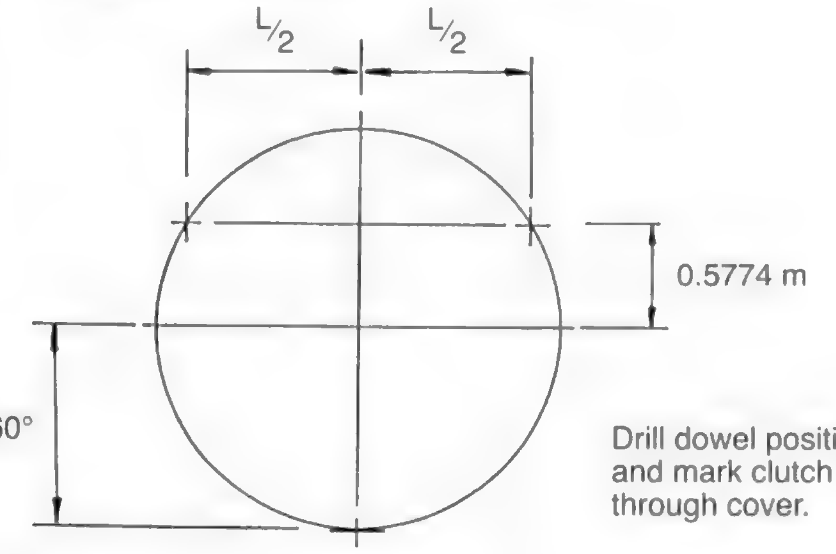
2l lightened rod
GC target weight after lightening, resizing, port-polish and shot-peen is approx 730gm, compared with approx 810gm standard, *ie* 80gm less. **(8/32)**

Crankshaft wedging
It is possible to remove metal from the crank webs of the TC cranks; the 2l example illustrated (for Tom Casey's Hot Rod) was machined by clamping the crank in an old block, with oversize bearings, and milling the webs (through the side of the block) with a multi-tip 3" dia facemill. A special depth gauge was fabricated to ensure accurate removal. Balancing was interesting(!), but only took John Woods about two hours. Final crank weighed 15.2kg, compared with 16.3kg standard [1kg off the crank (or flywheel) is worth about 20kg off the car].

8/33: DETERMINATION OF DOWEL POSITIONS (FROM CLUTCH COVER) WHEN FITTING NON-STANDARD CLUTCH.
Most clutch covers are centred on flywheel by means of 3 accurately drilled/reamed dowel holes equi-spaced at 120° as shown.
TO DETERMINE POSITIONS OF DOWELS FROM FW CENTRE:

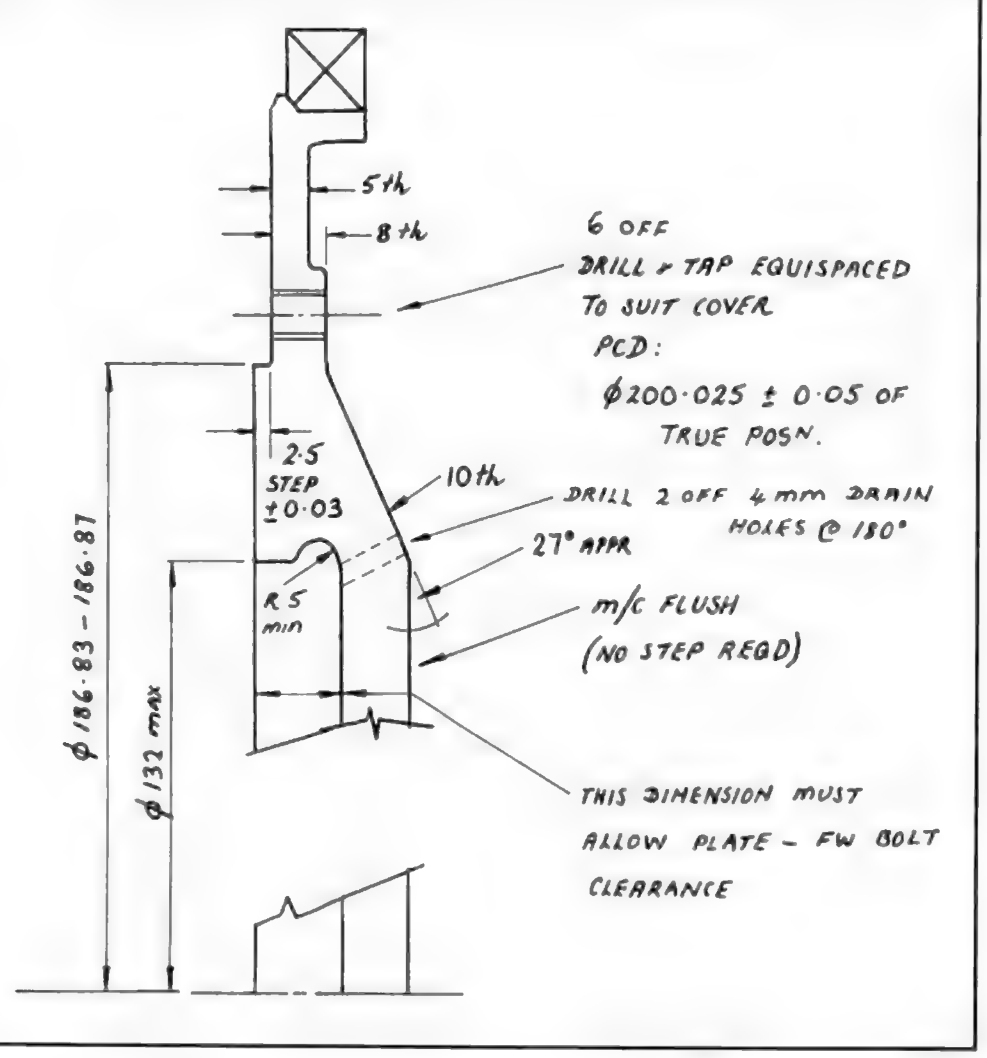
1. Measure dist L with a vernier calliper L ÷ 2 = m
2. Pitch circle radius of dowels r = m ÷ sin 60°
3. Dist h = r sin 30°= m sin 30° ÷ sin 60°
4. Hence h = m x 0.5774





Drill dowel positions as shown and mark clutch bolt positions through cover.

Flywheels (2/ n/a)
Standard flywheel weighs approximately 8.4kg.
GCT lightened version (for standard-type clutch) 7.1kg.
GCT ultra-light ¼" En8 race flywheel 4.4kg.
GCT ultra-light En8 flywheel to Gp N clutch 5.6kg.
(All weights include ring gear.)



8/34: STEEL FLYWHEEL TO SUIT SACHS / AP RACING 7¼" CLUTCH
(single or twin-plate)
Key dimensions and minimum thickness (mm)
Radius or blend all sharp corners to minimize stress concentrations
(PCD SHOWN FITS ALL 7¼" CLUTCHES)
Outer section (4mm thick) can be lightened by milling (see photo).

8/35: Incorrectly machined cast iron flywheel: machined far too thin across centre. This particular item landed neatly in the bellhousing of a Stratos replica at low revs!

8/36: When this grasstrack flywheel finally exploded, this was all that was left of clutch (Gp N Sachs). It was at about this time that GC fatigue-life schedule for cast iron flywheels had to be revised! No, they don't last forever: correctly lightened and balanced, this one did four seasons. Force of impact tore mounting lugs off block and bellhousing.

CHAPTER 9

PISTONS AND RINGS

The piston has two primary functions: to provide the necessary means of achieving the required compression ratio (via a bowl in the piston crown, or 'intruder' dome), and to allow the gas force in the cylinder to be transferred to the crankshaft via the con-rod. In performing its design function, the piston is subjected to complex effects of pressure, heat and acceleration, all the while being subjected to the forces generated by the friction in the cylinder of the rings and skirts.

The design of high-strength low-expansion diecast alloy pistons is a major contributor to the success of modern high-speed engines, and such is the quality of production pistons from the main manufacturers, AE (incorporating Hepolite and Borgo Nova), KS (Kolbenschmidt), Mahle, TRW, Mondial, that GCT have been able to employ them in applications well beyond their design limits.

From the equation:

$$CR = 1 + \frac{Vs}{Vc}$$

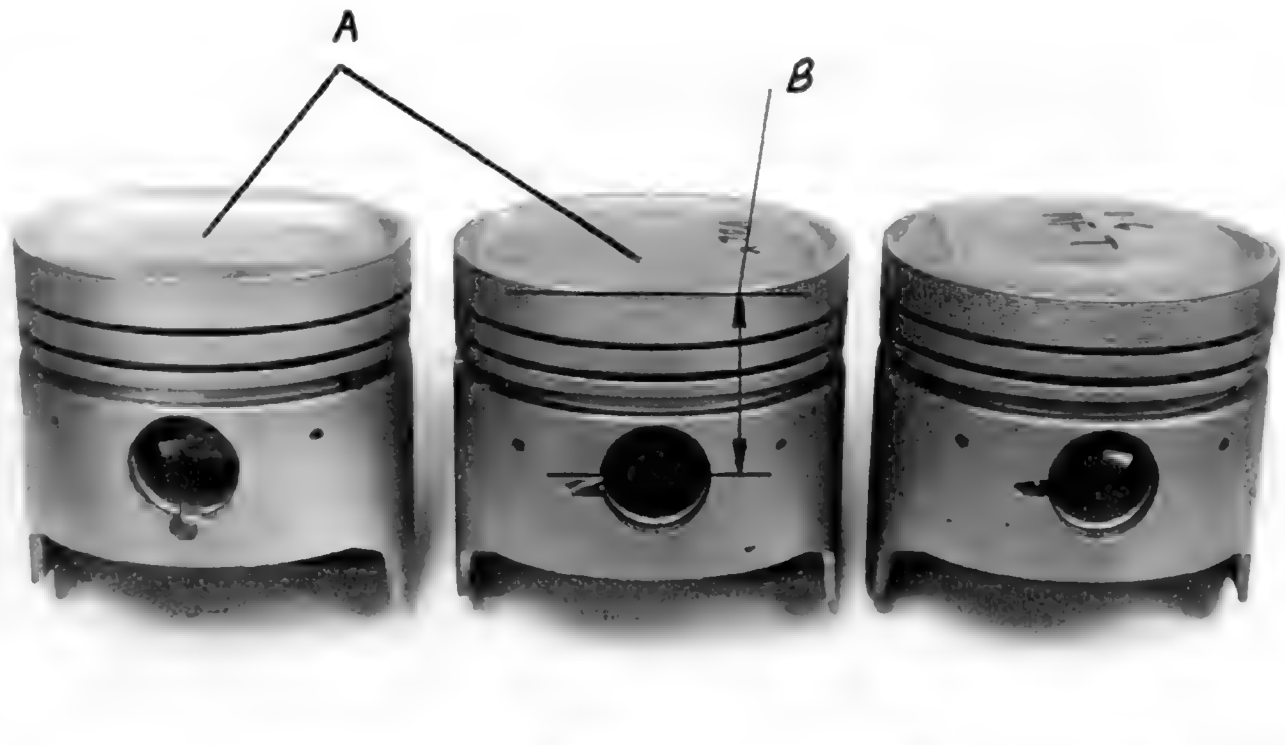
it is clear that the CR can be varied by piston changes, either in terms of the bore size (giving a larger value of Vs) or the design of the piston crown. To raise the CR, an 'intruder' dome can be used; to lower it a bowl can be incorporated into the crown, thus altering the value of Vc for a given combustion chamber volume and gasket thickness. US production models designed for low-octane fuel (95 RON) incorporate very large valve cutouts to achieve their lower CR. Where a dome is used it may be solid (cast pistons), or hollow in the case of forged; where the piston is bowled (*eg* 8v Integrale/Croma Turbo, Volumex) the piston crown is thicker to allow adequate strength. Above a certain limit, around 11:1 static CR, the conventional truncated cone dome design used on most forged types starts to upset the development of the flame front during combustion and this seriously affects the engine's thermal efficiency (more fuel per bhp is needed). This is an inescapable feature of the 8v TC. (9/1)

9/1: Flat-top Venolia forged pistons for Volumex with raised (8:1) CR, cutouts machined individually (note offset towards centre of engine on 1 and 4). Pistons are pin-punched through valve guide at TDC to mark valve centre and then cutouts machined on mill in tilting vice with flycutter.

Apart from this, the only finite limiting factor on dome height is the knock resistance of the fuel *(see Case History No 4)*. Remember that 'hotter' cams and a high-energy intake system raise the effective CR.

The influence of heat

In general, the temperature of the hottest part of the piston is the centre of the crown, which varies with engine load/speed between 150°C–350°C. Forged pistons run approximately 10–15% cooler due to the denser metal structure and high copper content. This is worth remembering when building a full-spec turbo motor, where the high temperatures increase the risk of



9/2: Three types of standard (diecast) pistons for, L–R, Croma Turbo, Volumex, Delta 1.6 Turbo (carb). Note thick land between first and second rings for pressure-charged applications. Section at A is referred to as 'well' or 'bowl'. Terminology: distance B from crown to centre of pin is compression height, which determines set-up height in cylinder.

9/3: AE press-fit pistons and rods for 125 Samantha. Fitting these types requires the con-rod to be heated. Pin is sliding fit in piston; when inserted and con-rod cools, small-end grips pin tightly. No pin clips needed. Terminology: piston dome is at A, crown is at B, skirt is at C. 124 Sport (BC) switched to full-floating pins. These 80mm-bore pistons are now virtually unobtainable – use forged.

detonation and consequent piston damage. A piston dissipates around 70% of the heat from the crown *via* the rings, lands and skirt to the cylinder walls, the rest being dispersed *via* the oil (and crankcase gases) and con-rod.

Because of the unusual configuration of temperature gradients around the piston (the hottest parts being nearest the top), production pistons are designed with oval skirts (viewed from the top) with a slight taper from the upper part of the skirt to the crown, *ie* viewed from the side. Viewed from the side, the skirts tend to be barrel-shaped. (9/2, 9/3)

At operating temperatures, a well-designed piston will expand so as to achieve a circular form to match the shape of the bore. For this reason, when pistons are being measured to check skirt-bore clearance, the micrometer should be held approximately 1cm from the bottom of the skirt, and the measurement of the largest diameter taken as the true reading.

From the 124 1608cc 80mm-bore onwards all the TCs used Autothermic pistons, whereby a steel strut is incorporated into the piston at the casting stage. This inhibits expansion of the piston skirts. Adoption of this design is now quite general in the motor industry, and contributes significantly to the long life of the engine since bore clearances can be minimized, and this cast-in strut design is considerably stronger than earlier split-skirt types.

Because the production pistons run with skirt-bore clearances of 0.05mm or less, it is vital that the appropriate bore size is used – too tight and the piston may seize. Additionally, there is a risk of scuffing in the area of the pin boss if an engine is started from cold and run immediately at high power, because a large amount of heat is conducted through this area and the piston expands more quickly than the bore – whilst the flexible skirts can cope with this, the rigid area of the pin boss cannot.

Excess heating of the pistons is most commonly caused by pre-ignition or detonation (owing to excessive air or engine temperature, incorrect fuelling etc). When this happens, the area of the piston nearest to the point of propagation usually starts to melt, with catastrophic results. A thick-section piston can tolerate a certain amount of scorching, but an ultra-light race piston may later suffer weakness and cracking, even though it appears to be structurally sound. If in doubt – replace. (9/4)

Piston acceleration

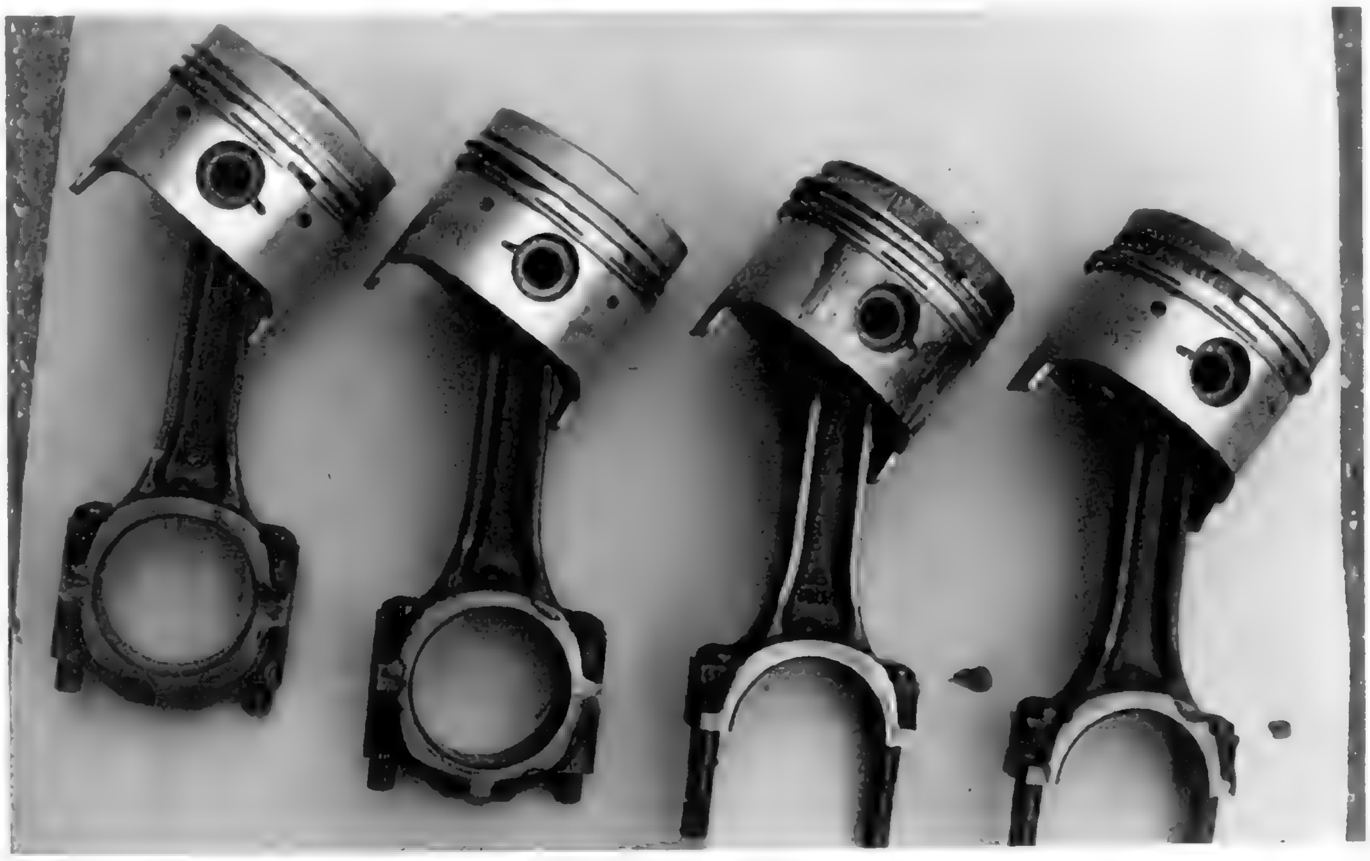
Maximum piston acceleration occurs at TDC and BDC and is given by the equation:

$$A = w^2R\left(1 + \frac{R}{L}\right)$$

where $w = \dfrac{2\pi \times rpm}{60}$ (radians/sec)

R = crank radius (*ie* ½ stroke) (ft)
L = rod length (ft)

Calculations of peak piston acceleration for the 2*l* and 1585cc models give the results reproduced in Table A on the next page:



9/4: Classic installation problem wrecked these AE 131 1600 pistons in Autograss car. New rules meant no external ducts on car, led to very high engine/intake temperature (creating excessively rich mixture) – result: detonation. Diagnosis was hampered by lack of proper instrumentation in car – a problem dealt with after the rebuild. How hot was it? Engine 105°C, underbonnet temperature not far off this. Pistons, block, bearings, guides and rings were wrecked in one race! Cold-air duct totally enclosing carbs cured detonation, but engine temperature problem was never really solved. High engine temperature robs power – see graph in Chapter 13.

TABLE A		
SPEED (rpm)	ACCELERATION (A) (ft/sec²)	
Engine	*2l*	1585cc
3000	19128	14494
4000	34005	25766
5000	53132	40260
6000	76511	57975
7000	104139	78910
8000	136019	103066
8500	153558	
9000		130443

It may be readily deduced, comparing for example the acceleration of the *2l* at 4000 and 8000rpm, that if the speed is doubled, the acceleration (and hence force) on the piston is multiplied by a factor of 4!

Note also that the acceleration of the 1585 is around 25% less than the *2l* for a given speed, due to its shorter stroke and rod. The fact that these two engines *will* withstand such high accelerations is certainly a tribute to the strength of the basic TC design – GCT have 'dynoed' safely at these speeds (with forged pistons)!

Since the product of mass × acceleration determines the inertia force on the pistons, rings and rods, it follows that stresses can be reduced by keeping the weight (or more correctly – mass) of these components to a minimum. A lightweight piston, rings and pin (the best race pistons use lightweight tapered pins) will reduce the load on the rod assembly, and at the same time reduce the inertia loss during the power stroke (when the gas pressure is trying to accelerate the assembly), leading to better torque output at the crank.

Pin-offset

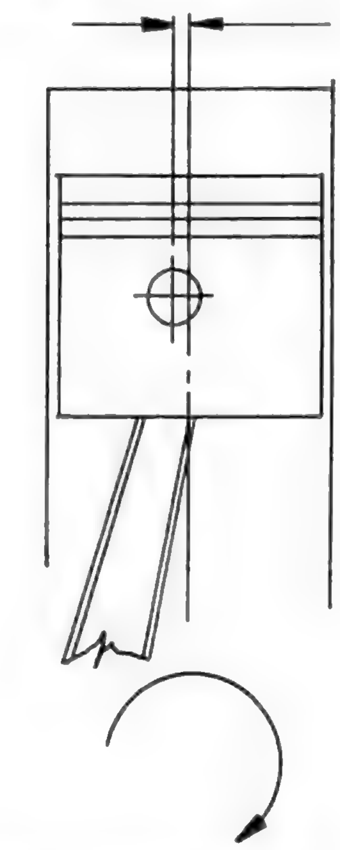
All production TC pistons have a pin-offset of 1mm or 2mm to reduce the tendency of the piston to 'slap' as the rod changes direction at TDC, thus reducing the force on the bores and piston skirts. Some manufacturers of forged pistons, *eg* Venolia, do not use this, in order to achieve a more direct 'push' on the piston. Clearly, it is important to ensure that, where a piston *does* have a pin, or 'thrust' offset, they are installed the correct way round in the bore. With n/a engines, this is straightforward since the position of the pistons is readily identifiable from the valve cutout shape; on pistons from models such as the Croma Turbo and other 8v turbo models – check carefully. Some manufacturers, *eg* Mahle, mark the pistons with a small diagram indicating the direction of rotation (9/6).

Piston ring design (9/7)

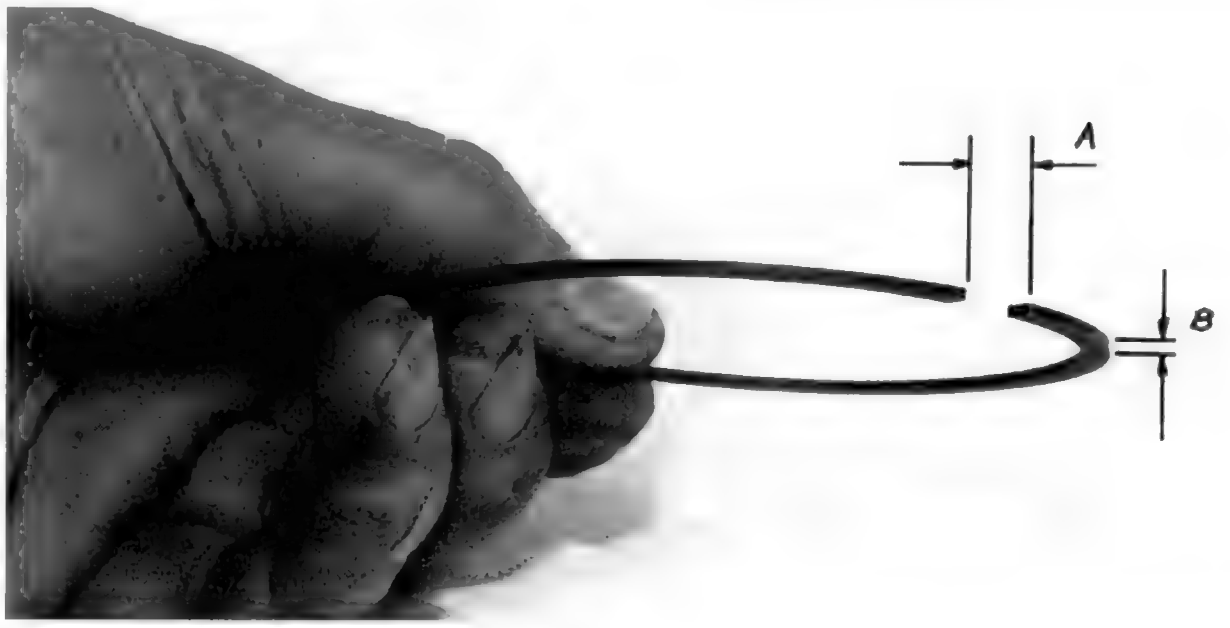
The radial force exerted on the bores by the rings depends on:

1 Strength of the ring material (cast iron or steel)

9/6 Piston pin offset to left of bore centreline.



Direction of crank rotation (clockwise on all TCs viewed from crank nose).



9/7: Ring data. End gap or 'fitted gap' is the distance A between the ends when installed in the bore. In the free state this is known as the 'free joint gap'. B is ring thickness (confusingly sometimes referred to as width).

2 Radial width
3 Free joint gap
4 Gas pressure and ring thickness

Chrome-plated steel top rings have been extensively adopted due to the greatly extended wear-life compared with cast iron. Additionally, the rubbing effect of the rings in the bores deposits minute quantities of chromium on the bore, helping to protect it from corrosion and abrasion. Phosphated or molybdenum-coated oil scraper/control rings are also used for the same reason, indeed some Turbo models (*eg* 16v) have even adopted a chrome plated steel second ring.

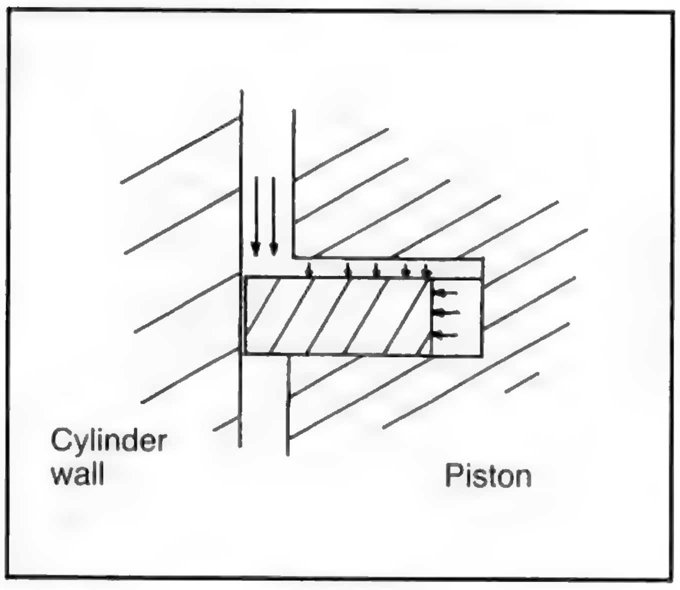
9/5: SOHC piston at left shows classic detonation damage due to over-lean mixture, high inlet temperature (no cool-air duct). Metal goes 'pasty' and is dragged into bores. Diecast piston at right was destroyed by high revs. It is not only speed that matters, but also duration of overspeed. GCT have seen cast pistons briefly revved to 10,000 in 2l and stay together!

A thick top ring will exert more compression force against the cylinder than a thin ring because of the greater area over which the cylinder pressure is distributed. The preference on race engines, however, is for thin, lightweight rings which reduce the radial force (hence friction) and ring inertia – helping to keep the rings on their seats at high rpm.

Piston ring performance

All TCs are fitted with three rings per piston. The purpose of the top ring (compression ring) is to seal the bore during compression and the firing stroke. The other two rings perform a secondary function of assisting with bore sealing, but their main task is to distribute and regulate lubricating oil in the bores and in doing so return it to the sump *via* drillings in the bottom ring groove.

Ideally, the top ring should remain in contact with the lower side of its groove during the compression and firing stroke. When the force of gas pressure in the cylinder acting on this ring is less than the inertia force generated by the effect of the ring mass and its upward acceleration, the ring will lift off its seat and its performance will be impaired. This is because the compression ring relies on the gas pressure acting as shown to seal effectively. (9/8)



9/8: Optimum performance of compression ring relies on gas pressure acting as shown.

The tendency of the top ring to come off its seat at high rpm (ring 'flutter') can be readily examined since the acceleration imparted to it, at various speeds, is known *(Table A)*. As the engine fires before TDC, gas pressure is building up as the piston ring achieves its maximum acceleration. The upward (inertia) force $F\uparrow$ on the ring is given by:

$F\uparrow$ = ring mass (lb) × A

and the downward force is given by:

$F\downarrow$ = peak cylinder pressure (lbf/in^2) × ring area (in^2)

where the ring area is the area of its top face.

The peak cylinder pressure, depending on CR, cam type, etc, may be in excess of 1200lbf/in^2, although this diminishes as the piston moves down the bore. Taking a typical top ring (production rings of cast iron and steel weigh roughly the same) as having a mass of 0.022lb, and a top surface area of 1.52in^2 (84mm bore), the value of the downward gas force $F\downarrow$ for various values of peak cylinder pressure is shown below.

TABLE B

PEAK CYLINDER PRESSURE lbf/in^2	$F\downarrow$ (lbf)
600	910
700	1062
800	1214
900	1365
1000	1517
1100	1669
1200	1820 †
1300	1972

† assumed peak pressure value of $F\downarrow$

The upward force $F\uparrow$ can be calculated to give: (neglecting friction)

TABLE C

SPEED (rpm)	$F\uparrow$ (lbf) (2*l*)	$F\uparrow$ (lbf) (1585)
3000	422	319
4000	750	568
5000	1171	887
6000	1686	1278
7000		
8000	2295	1739
8500	3384	
9000		2875

If, for example, the peak pressure is assumed as 1200lbf/in^2, it can be seen that $F\downarrow$ (1820lbf/in^2), for the 2*l* is less than $F\uparrow$ at about 6500rpm, and for 1585 at around 7500rpm, so it can be seen that even at this high pressure, the top ring must be lifting off its seat at these speeds. When this happens, the gas cannot get behind the ring and sealing force must be lost. This is one reason why torque can fall off at high rpm.

These results are more or less in line with the industry standard that the maximum acceleration for a 1.5mm thick ring should not be greater than 80,000ft/sec^2. Lightweight rings will show a power advantage.

Ring lands

The piston ring lands must be strong enough to withstand the explosion pressure and should be located around 1/10th of piston diameter (mm) below the crown. Turbo and supercharged cast pistons have thicker lands than their n/a counterparts, but forged types, being far stronger, can be designed with the same land thickness as n/a engines up to 30lbf/in^2 boost. The real killer, as far as the land between the first and second ring is concerned, is detonation, which tends to cause the land to crack, or in extreme cases, fracture altogether.

On n/a race engines, cast pistons should not be used above CR 10.8:1 as there is a danger of the high peak cylinder pressure leading to the same result.

PISTON INSPECTION – DIECAST PISTONS

The most crucial checks on pistons are:

1 Cracks or damage
2 Skirt size
3 Pin fit
4 Rings, grooves and lands
5 Ring end gap

Cracks

The easiest way to check for cracks is to thoroughly clean the pistons and inspect visually (if necessary with a magnifying glass). Beadblasting is highly effective, but shield the ring grooves with tape and ensure that no media is trapped in the pin clip recesses afterwards. Solvent and Scotchbrite are also good for cleaning the skirts, lands and crown, but not much use for the underside due to its complex shape! Cracks will tend to exist:

1 Around the valve cutout
2 In the centre of the bowl (to boosted engines only)
3 In the ring grooves or lands
4 At the junction of the pin boss with the skirts

Unless the history of the pistons is known, do not be tempted to use secondhand ones; their fatigue strength may be drastically low due to hard use.

Dye penetrant crack detection can obviously be used, but this is not practical on complex areas of the piston as the dye penetrant has to be completely wiped off before the developer is applied. If cracks are present, the dye trapped in them will be drawn into the developer to show their position – if surface dye is present when the developer is applied it will give a false reading.

If there is any doubt as to the history of a set of used pistons, forged or cast – replace.

Check the pistons for impact damage with the head or valves – light damage can be dressed out with a die grinder and carborundum paper.

Size

Measure the skirt at its widest point, usually approximately 1cm from the base, and compare with piston data. Late (84mm) production pistons (including Mahle, KS, etc) run with 0.04–0.06mm skirt-bore clearance. This may be increased to 0.075 (GC tests) without noticeable penalty – or advantage – if needed. Too much clearance will lead to piston slap, which in extreme cases can snap the edge of the valve cutout; too little and the piston may seize. The minimum stated clearance must not be reduced by more than 0.005mm.

Remember that production pistons and bores are size-classed, *eg* for the 131 2*l*

Class A piston dia = 83.950 – 83.960mm
" B " " = 83.960 – 83.970mm
" C " " = 83.970 – 83.980mm
" D " " = 83.980 – 83.990mm
" E " " = 83.990 – 84.000mm

The result of this is that, for example, a Class E piston of 83.990mm size cannot be used in a Class A bore as the Class A bore size even at a maximum of 83.960 plus clearance, *ie* 84.01mm, would only allow 0.02mm running clearance. Although this comment seems superfluous (as most owners would rebore and fit oversize pistons), it is relevant in the case of a piston-swap between two different blocks (*eg* 1585 Beta to 2*l*) whether new or used. Obviously, if such a swap of production pistons is desirable from a cost point of view, the bores must be honed to suit. If the skirts are worn undersize more than 0.025mm, consider new pistons.

Examine the skirts for scuffing – from carbon impact damage, bore washing (lack of lubricant caused by excessively rich mixture) or overheating. Minor damage can be dressed out with 400-grade and oil. Clean with 'trike' to remove all traces of abrasive afterwards.

Pin fit

The pins on production pistons are sized to their respective pistons and should not be interchanged. The fit clearance between the pin and its bore is 1/10thou" – 3/10thou". Pins rarely wear, but a worn bore can cause impact loading on the pin boss and noise. When new and clean (lightly oiled), fully-floating pins should be a finger-push fit in the pistons. The piston bores go 'oval' in service; this is quite normal and no remedial action should be taken to loosen the pin fit.

Rings, grooves and lands

Rings and grooves wear progressively during service. Their standard dimensions and clearances need to be determined from data from the individual piston manufacturer. For example, the following dimensions are quoted by Fiat for most models, including 131, 130 TC and HF Integrale:

	GROOVE (mm)	RING THICKNESS (mm)	SIDE CLEARANCE (thou")
TOP	1.535–1.555	1.478–1.490	2–3
CENTRE	2.030–2.050	1.980–2.000	1–3
BOTTOM	3.967–3.987	3.925–3.937	1–2½

Once the ring or groove becomes worn and the clearance increases by more than around 40% of the above, replacement will be necessary. Wear in this area leads to compression loss, bore wear and high oil consumption. Check the side clearance with a feeler gauge.

Check the condition of the piston lands for cracks most carefully – cracks can be easily overlooked. As there is a generous clearance between the lands and bores (to allow for expansion), minor impact damage can be dressed out as described earlier.

Ring end gaps

TCs are built with the following end gaps (84mm bore):

early turbos and normally aspirated
top 12–18 thou"
centre 12–18 thou"
bottom 10–16 thou"
late turbos (*eg* Integrale):
top and
centre ring 12–20 thou"

As the same ring sets are used on the various size Classes A–E, Class A will have the tightest fit. (GCT normally aim for 15thou" top and centre ring end gaps on rebuilt n/a engines.)

An excessively high end gap leads to loss of compression. A well used St III or full-race n/a engine should only exhibit around 20thou" top ring end gap at the end of the season. This is the limit for a wet-sump engine: a dry-sump unit can tolerate around 22thou" as the crankcase gas is evacuated by the pump – diminishing the risk of oil loss due to crankcase pressurization, although there may be as much as 10% power loss due to leakage over an engine with new (run-in) rings. (Flex-Honing greatly increases ring/bore life over conventional honing.) Ring end gaps on used or new rings can be readily measured by fitting them in the bore and using a feeler gauge.

PISTON INSPECTION – FORGED PISTONS

Forged pistons differ primarily in that they have different expansion rates than diecast types, and skirt-bore clearances will be correspondingly larger. This is due to the lower silicon content (silicon inhibits expansion) and a higher concentration of copper and magnesium, plus the possible addition of titanium. Forged pistons may have anything from 2–7thou" skirt-bore clearance and also looser fitting pin pins/rings. Some high-expansion types (especially US) can give rise to audible piston slap when cold – this disappears as the piston expands in the bore. Checking requires data from the manufacturer.

Piston pin retention

All diecast pistons use wire clips for holding the pin in place. Clips should be renewed if the pistons are being used, although, provided the fitting tool described in *Chapter 3* is used, GCT have had no problems with re-using old clips – provided that they are opened out first to ensure adequate preload in their groove. Retention with forged types may be a wire clip or Teflon button. Teflon buttons must always be renewed as they are prone to wear, but are preferred for very high rpm use because of the tendency of clips to 'jump out' at TDC due to the very high piston acceleration.

Selecting pistons

One great advantage of the 84mm-bore TC engines is that a high compression

ratio can be achieved by substituting pistons from another model. As outlined earlier, a high CR leads directly to more power (and better thermal efficiency) and may be vital to ensure that the best possible torque is extracted from a cam change. That said, it must be remembered that changing pistons is not a simple matter of taking one set out of an old engine and putting them in another block *(see later)*!

The following table gives an approximate idea of the CR resulting from a piston change:

9/10: A 16v piston (Venolia) designed by GCT for converting 131 block to 16v with Thema 16v head. Minor combustion chamber relief needed. CR over 11.5:1.

PISTON FROM	IN ENGINE	GIVES APPROXIMATE STATIC CR
131 1600	2*l* 131/132	10:1
132 (GLS-type) 1800	2*l* 131/132	9.5:1
Beta 1585	2*l* 131/132	10.8:1
Beta 1585	131 1600	9.5:1
Beta 1585	132 1800	9.8:1

Notes: Pistons from the 105TC are similar to the Beta 1600 with a slightly taller dome. Pistons from the 124 Sport 1800 (9.8:1 CR version) are similar also to the Beta 1600 type.

RPM limits

GCT have experienced no problems with piston failure by adherence to the following rpm limits *(see also life schedules – Appendix A)*

STROKE (mm)	TYPICAL ENGINE	CAST PISTONS MAX SPEED (rpm)	FORGED PISTONS
71.5	1585 131, Beta, 105TC	8500	9500
79.2	1800, 124, 132	8000	9000*
90	2*l*	7200	8600

* GC Mk 1 Hydroplane was actually raced at 10,500rpm at Lake Windermere! The 'killer' is not so much the engine speed itself as the *time* for which the engine is held at high speed. (9/9, 9/10)

Bore size

Increasing the bore size will, theoretically, give more power because of the increased engine size *per se*; the CR may not necessarily increase because manufacturers tend to reduce the height of oversize pistons to keep the CR (with increased Vs) the same as the Fiat/Lancia original specification. GCT do not have any data to quantify bore size/power increase, but taking, for example, the 2*l* from 84mm bore (1995cc) to 85mm (2049cc) would certainly lead to a measurable power increase, even with the same CR.

Generally (unless the engine is being tuned to the limit, where 'every little counts'), it is advisable to stick to the smallest rebore size at which the block will 'clean up'. Oversize cast pistons are generally made by Fiat/Lancia at +0.4, 0.6mm and other manufacturers also at +0.8, +1mm, but check before specifying a rebore size that the dome configuration to give the required CR is available. AE 131 1600 pistons, for example (commonly used by GCT on St II 2*l* engines to give around 10:1), are only available at 84mm, +0.4 and +0.6 oversize.

Gp A forged pistons are usually available 'off the shelf' from Mondial and Asso Werke (Italy) in fixed sizes. If ordering forged pistons from Venolia (US) or UK-based Cosworth, Holbay, Omega, Arias or Accralite, liaise closely with the supplier over bore size, dome/crown configuration and ring size (GCT have always used production ringsets for the 131 2*l* from AE, TRW, Mahle or Kolben Schmidt, although various designs of lightweight race rings are available). When specifying a large oversize, *eg* 86mm (which gives approximately 2.1*l* on the 2*l*) the 128 1300 rings can conveniently be used. (Ringsets are sized by width and bore diameter.)

Calculating bore size

The final cubic capacity of the engine may be calculated simply from the equation:

cubic capacity (cc)

$$= \text{no of cyls} \times \text{swept volume (cc)}$$

$$= 4 \times \frac{\pi D^2}{4000} \times S$$

where D = bore size (mm)
S = stroke (mm)

9/9: Venolia forged pistons for HC 8v TC. GC types are designed to accept standard ringset. These pistons have no thrust offset. Forging gives far stronger piston than cast – essential for high-rpm operation and better heat dispersion – and are particularly useful for high-boost turbos.

Rebore, re-ring or sleeve?

Donor engines with over 30,000 miles on the clock, or which have suffered oil neglect (or incorrect mixture) may have a significant 'wear ridge' at the top of the bores. Rings used to be available with a step (RD ring – or 'ridge dodger') to ensure that, if new rings were fitted, they

did not clip this ridge, causing damage to the compression ring. These RD rings are now hard to get.

The question of what to do with a block, in terms of reconditioning, is largely down to a number of factors:

a Are the pistons worth saving? (especially forged items)
b Are RD rings available?
c What will be the cost of 'dressing out' the wear ridge, and will the final bore size (after deglazing) be suitable for the pistons and rings?
d What is the bore size upper limit for the competition bore class size? (Some race regulations have very strict size limits.)
e If using original size – Classes A–E pistons – what will the cost be of honing the bores to suit?
f Will a rebore be cheaper (and quicker)?
g On a fully prepared block, is it worth sleeving one or more bores to save the block and re-use the pistons?

Because of the extent of the work involved in re-ringing, GCT have always preferred to rebore, except where, for example, a good used race engine has allowable skirt/bore wear (where original size rings will give suitable end-gaps, or oversize rings can be 'gapped down' to fit) and can be merely deglazed. Re-ringing a badly worn block is not really viable because it takes so long, and excessive skirt/bore clearances may result, leading to piston slap and possible damage to the top lands and valve cutout edges (oversize rings would be used and 'gapped-down' to suit the bore – see below).

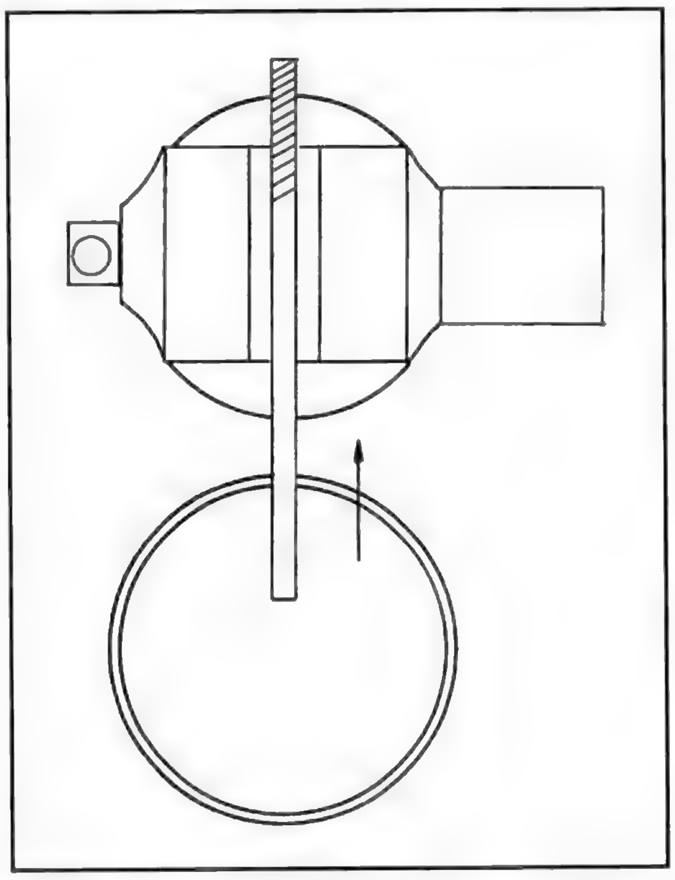
Sleeving is only worthwhile where the engine is fitted with forged pistons, or where only one or two bores have gone over the limit of running clearance or class size, and oversize pistons are not available or permissible.

If fitting used pistons, clean the crown and exterior of the piston with 400-grade carborundum paper (wet) or medium-grade Scotchbrite to remove excess carbon and dress-out minor scratches. Used pistons very often have minor impact damage above the top ring, where carbon has been impacted. Clean the ring grooves with an old broken ring and then dress the contact faces of the grooves with 1000-grade. Beadblasting old pistons is OK, but tape off the ring grooves first. After beadblasting, roughen the skirts with 400-grade or Scotchbrite to restore the oil retention and make sure all traces of media are thoroughly cleaned off. Only beadblast pistons with fine media, *eg* Guysons Honite, or impact damage may occur.

Ring gapping

In the absence of a specialist ring gapping tool, this procedure may be satisfactorily carried out by clamping a fine file in the vice and working the rings along it, holding the ring firmly to ensure equal metal removal from both sides. It is important, with chrome or moly-faced rings, not to chip the contact face of the ring, so only cut in the direction shown. (9/11)

Remove a small amount, lightly dress the outer edges with a stone or 1000-grade (working inwards), check the rings in the bores, and finally dress the edges of the ring carefully with a fine carborundum stone or 1000-grade wrapped round a thin file.



9/11: Fine file clamped in vice can be used to gap rings. Work ring along file in direction shown while exerting even hand pressure on each side.

It is not advisable to use rings more than +0.2mm larger than the nominal bore size for this procedure, *eg* use 84.6 rings in an 84.4mm bore. Be gentle with the rings during regapping so as not to spoil their tension.

Setting-up pistons

This is a question of dome configuration, crown height and cutout size. Piston crowns can be modified in a lathe so that a piston excessively proud in the block can be reduced in height. GCT recommend 20thou" up as the maximum. Certainly gasket thickness is a factor, but always have 25thou" clearance between the piston (dome or crown) and head to allow for rod stretch, expansion, etc. The piston dome, although it intrudes into the flamefront, is the main contributor to a high CR, and hence torque, and for ultimate results may be very large indeed!

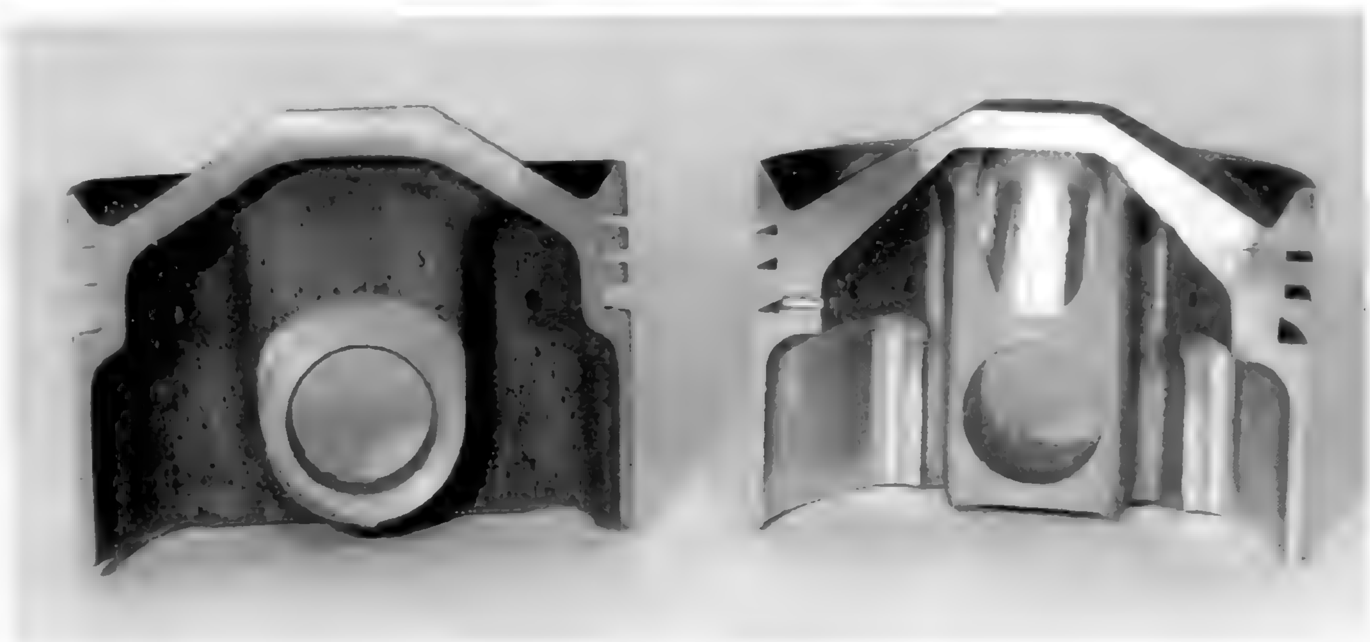
Consequently, on 84mm+ bore engines the combustion chamber may need to be opened out (especially at the front of No 1 chamber and rear of No 4, due to the combustion chamber offset – check clearance with Plasticine when dry-building) to suit the pistons. If the dome has to be modified – make sure enough thickness remains. Generally, manufacturers of forged pistons produce a truncated cone shape for the dome. If a special design is required, discuss this with the manufacturer at an early stage – hollow-dome (lightweight) forgings may not be suitable. (9/12)

Cutouts

See *Dry-building*.

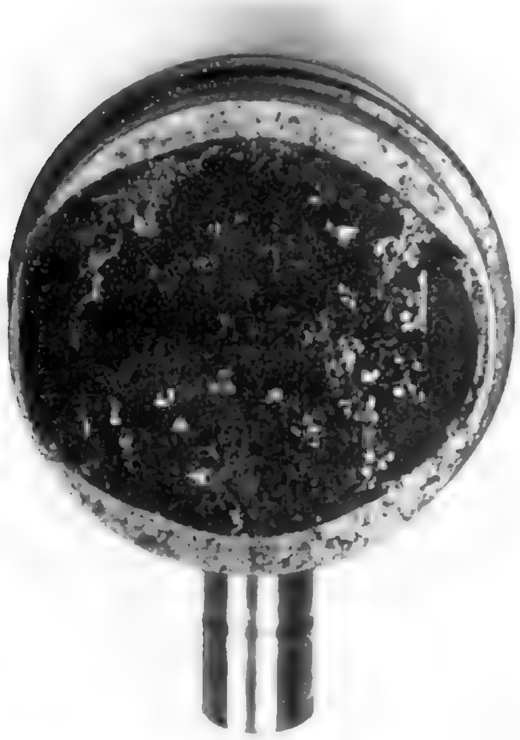
Piston repairs

Light damage to pistons may be repaired by welding, but this is only really feasible



9/12: Mondial (L) and Venolia (R) forged pistons. Note difference in design of arch under piston crown and relative thickness of skirts compared with pin boss area. Skirt must be flexible to cope with warm-up phase. Available metal between valve cutout and top ring can be critical. Dome thickness can be run as low as 3mm on normally aspirated units (eg NHRA engine – Case History No 10). These are hollow-dome forgings.

in the centre area of the dome/crown due to the risk of distortion. Do not attempt to repair pistons damaged by detonation because the whole microstructure in the area of the damage may have been weakened. Check on the composition of the piston alloy on an old piston before welding (by Tig process) because a high zinc or silicon content may produce a weld with excessive porosity. Never attempt repairs on other parts of the piston.



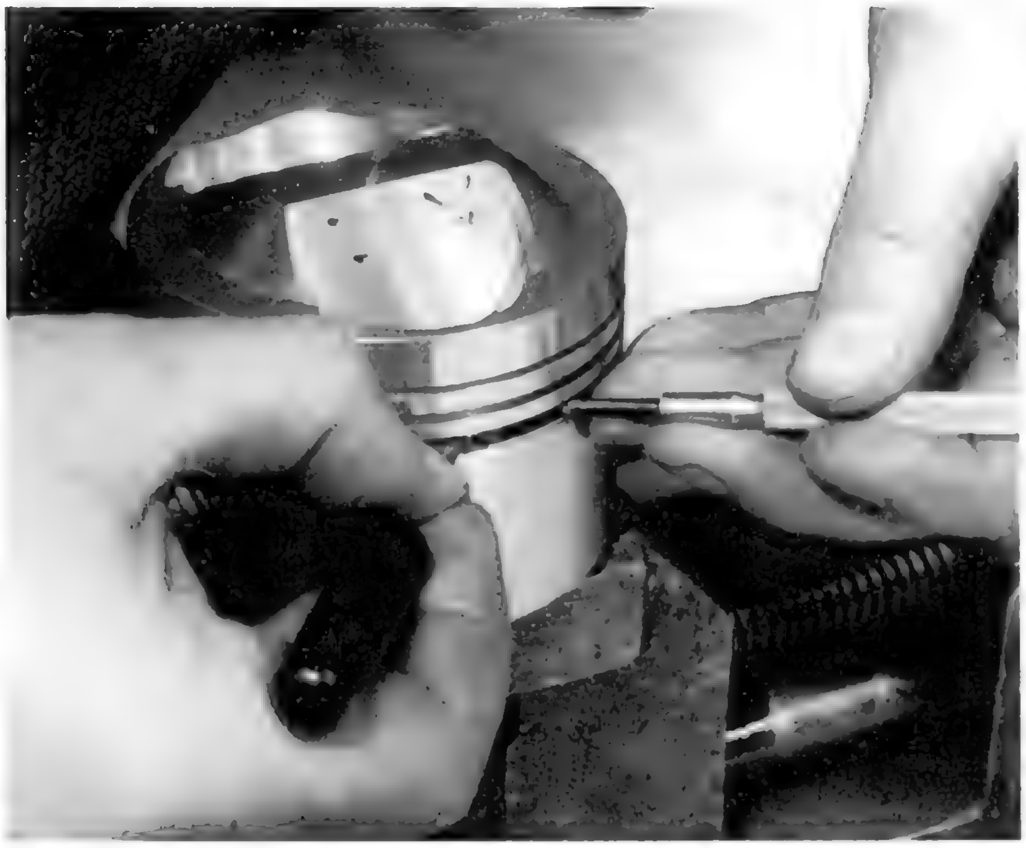
9/13: Damage caused by piston hitting head. Always allow at least 25thou" between piston and head to allow for rod stretch, crank warp at high revs. Beware! If pistons hit head at high rpm, piston pin clips may come out!

9/14: Inadequate dome-cylinder head clearance caused damage on this Venolia piston. Owner must have thought noise would go away on its own! Detonation damage to top land intruded into ring groove also – piston is scrap. (Engine had wrong emulsion tubes – too lean.) A silly way to wreck an expensive set of race pistons.

9/15: Fitting pin clips on forged pistons: mind your eyes! Most cast types have heavier clips and special tool is advisable if damage to clips is to be avoided.

9/16: Prior to fitting – lubricate rods, pins and pistons. 1608 124 BC and all late TCs have 'fully floating' pins which are lubricated from oil returning through holes in oil control ring groove to inside of piston, or in this case, via special 'pin oiler' drillings in piston boss fed from holes visible (Venolia type). Use of molybdenum or graphite grease helps bedding-in of skirts and rings.

9/17: Fitting three-piece lower ring (1). Fit expander ring: ends must butt together, not overlap. This type of ring relies on circumferential pressure from expander ring to achieve good contact between rails and bores. These 'latest spec' oil control rings are available from AE and TRW and give best radial pressure of any type.



9/18: Fitting three-piece lower ring (2). Fit upper and lower rails of bottom ring. Flexible steel rail sections are easily fitted spirally by hand. After fitting, ensure ends of rings are spaced 2cm either side of expander butt joint.

9/19: Second ring is cast-iron on most sets. Ensure it is fitted right way up – markings always uppermost on all rings. Use ring expander here (Snap-On tool). 16v Integrale uses chrome steel second ring. Be careful with cast iron rings – very brittle.

9/20: Another type of ring expander in use. Top ring here is chrome plated steel. Chrome top rings require 180-grit hone finish, cast-iron and moly faced cast-iron 220-grit. (Snap-On tool.)

9/21: Forged Venolia piston at left damaged by dropped valve caused by stones under cam belt on Hot Rod. Venolia piston at right killed in race hydroplane by inferior oil. Bearing seized, rod broke, goodbye engine! Note that despite damage, basic piston structures are still intact. Forged pistons are easily damaged: if using them make sure the rest of the engine is 'spot-on'!

CASE HISTORY No 4

Owner	Jonathan Douglas
Type	131 1600
Use	Race Morgan
Tested	JE Engineering, Coventry
Rig	Froude Consine

Specification:
CR 13.7:1 (see below).
44/38 valves.
Forged Volcanic pistons.
Bore 86.8mm (1692cc).
4-1 exhaust.
GC IIIA cams.
Fully mapped ignition.
Ford V6 Essex carb 40 DFAV.
Triple springs, alloy caps, copper head gasket, wired block.
Fully heat-treated dowelled crank.
Ultra-lightweight steel flywheel.
Lightened, polished, shot-peened, balanced rods.
Supergreen (high-octane unleaded) fuel.

This engine was built to examine the ultimate output from a single twin-choke downdraught carburettor, hence the high CR, on a big-bore (*circa* 1700cc) 1600 131. Owing to the risk of detonation, mapped ignition was deemed essential and the engine was run for mapping purposes on the detonation limit throughout the full range of engine speeds and throttle positions. Outstanding results were achieved, despite the two main problems of mixture distribution (each cylinder was monitored by Lambda-Sond sensors in the exhaust primaries) with the inlet manifold and the tendency of the carburettor to run lean at high revs (wide-open throttle). Neither of these problems was comprehensively cured: the carburettor high-speed enrichment circuit was enlarged as far as possible, but despite modifications to the inlet manifold it was felt that only a complete redesign with equal-length tracts would provide the definitive induction.

Volcanic piston design (CH4/1)

Pistons were supplied with a solid dome 12mm high, 85mm diameter by Venolia. This allowed the necessary machining on the dome to be carried out to achieve the very high CR. A figure of 13.7:1 was chosen to give a CR of 9.5:1 calculated from the effective stroke length at inlet valve closure, *ie* 78deg after BDC. The cylinder head was welded up to enlarge and equalize the squish bands adjacent to the spark plugs, and the opposite side of the chambers was enlarged using the die grinder to achieve a perfect hemispherical shape.

The idea was to fill the combustion chamber completely, leaving a minimum of 'end gas' at the rear of the chamber. A miniature Volcanic combustion chamber was created directly under the plug by machining the piston dome with a large ball cutter. The front of the piston was machined with a slot drill to accommodate the squish band, and the dome was carefully machined in the lathe to match the hemispherical shape of the chamber.

Finally, the camshaft lift characteristics were measured between (inlet) TDC-20deg ATDC, (exhaust) 20deg BTDC-TDC to assess the cutout depth required because it needed to be the minimum to achieve a vertical clearance between valve and piston of 100thou". Interestingly, the closest proximity occurred at (inlet) 18deg ATDC, (exhaust) 18deg BTDC, which emphasises the importance of dry-building – measurement of vertical clearance 'around TDC' is not sufficiently accurate.

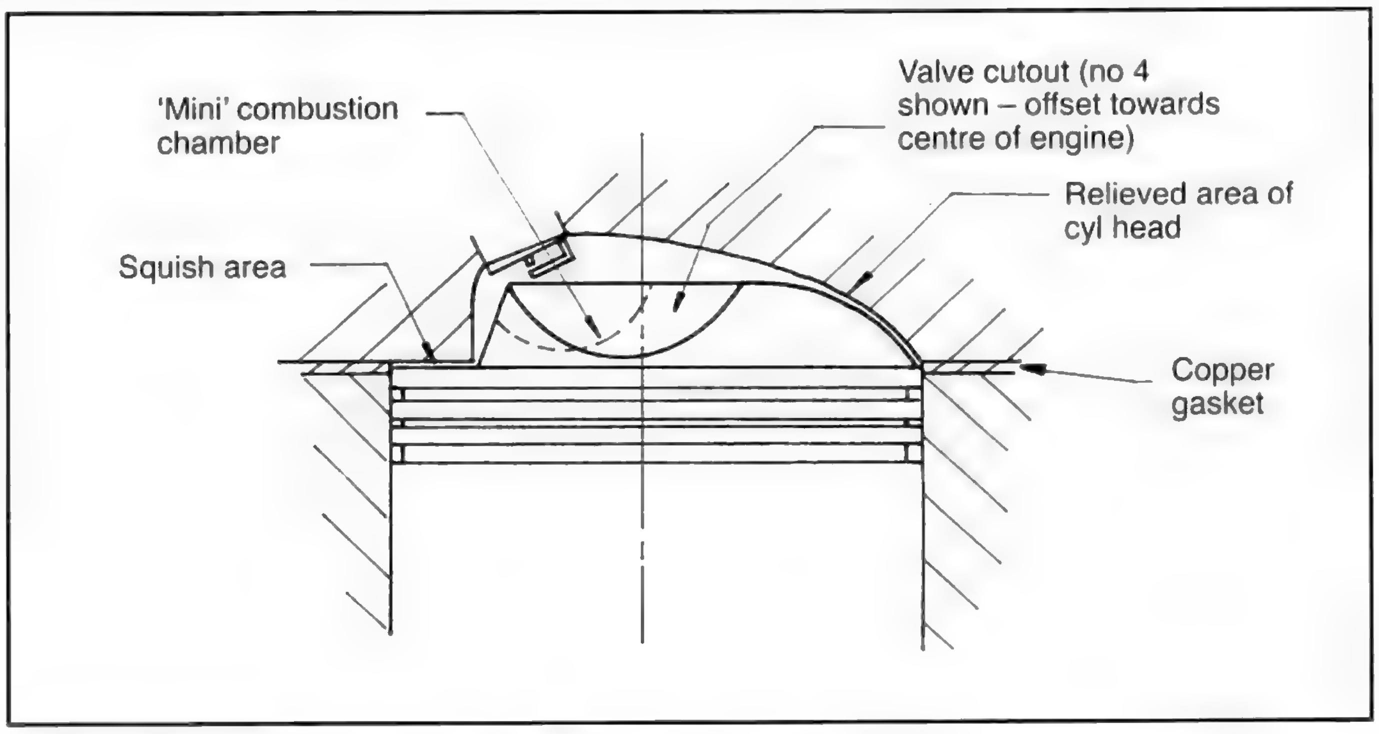
A minimum chamber wall thickness of 3mm was considered about right, but owing to the positioning of the coolant gallery at the front of the head (directly below the coolant outlet elbow), which was not, unfortunately, allowed for, the chamber burst during testing, necessitating a weld repair. (Two cylinder heads were sectioned prior to manufacture of the finished head.)

The piston dome was increased in height by machining 2mm off the piston crown; it was felt that this would not unduly reduce the strength of the top land. The block face was machined 2.5mm and the pistons run 20thou" proud of the block face (the compressed thickness of the copper gasket was 60thou").

After approximately four hours of load testing a minor coolant leak developed from the inlet side of the block between the head gasket and block. This was traced to the butt-jointed stainless wire ring inserts. The tiny gap (barely visible) was allowing gas to bleed under the gasket, pressurizing the cooling circuit. On the original build, Hermetite Gold sealant had been used on the block face. The rebuild (on the dyno) with a new annealed gasket used Silicon RTV instead, and a clear ring 5mm wide was left around the inserts to ensure that the gasket seated directly on the block. This cured the problem, but it was felt that in future the wire should be either copper or silver and the ends butt-jointed and silver-soldered, or overleaved.

After torque-up, the copper gaskets work-hardened and re-torque was not required. The crank chopper was machined from billet steel and a toothed belt alternator/water pump drive was used with a reduction in speed to 30% of crank speed; the 131 2*l* water pump performed perfectly. Preliminary results were very encouraging, showing the engine to be capable of around 124lbf ft torque, pulling strongly from around 4000rpm with quite a flat torque characteristic and with over 150bhp at 8000.

Jonathan Douglas commented shortly afterwards: '...The performance range of the single carburettor is as much a problem as the inherent problems of the manifold...The present situation is that I have had occasional bouts of running it on the dyno, but recently it broke again – either the gasket or the head itself...I've tried all sorts of amazing modifications to the carb, but it just can't cope. I think I shall take some time to redesign some of the internal passages of the carb to improve the volume of all the passages along which the fuel has to flow, and possibly try a lower fuel level to reduce lower-end richness...'



CH4/1: Diagram showing design principles of Volcanic piston.

CHAPTER 10

FUEL SYSTEMS

PART ONE: CARBURETTOR SYSTEMS

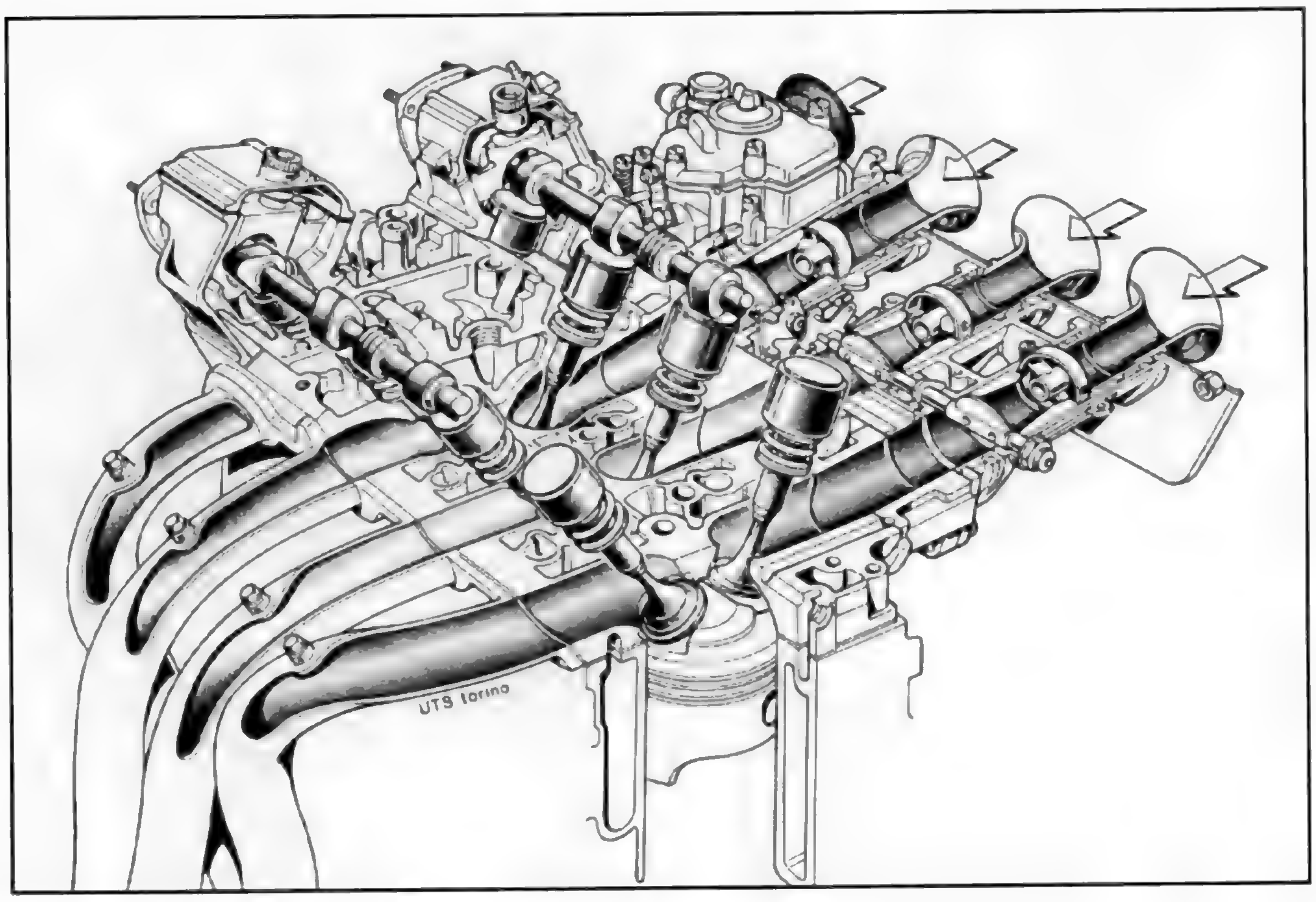
The normally aspirated TC responds superbly to the fitment of twin carburettors – always providing that the calibration and sizing are correct – and it was no coincidence that the launch of the 130 TC, with its strong and flat torque curve, was greeted with rave reviews by the motoring press in the mid-80s. (**10/1**)

Twin carbs (DCOE and DHLA, IDF and DRLA or DCNF) do not have any junction between the inlet tracts of adjacent cylinders, which means that there is no interference between cylinders caused by conflicting pressure waves. When an inlet valve opens, the sudden depression created at the valve throat causes a negative wave (depression) to travel up the inlet tract. For a given swept volume its magnitude varies according to the exact inlet port layout, valve size, engine speed and camshaft configuration.

As this depression reaches the rampipe mouth (on full throttle), if the inlet tract length is compatible with the wave frequency it will be reflected as a positive (pressure) wave at certain speeds, which helps to force the charge into the cylinder. This wave can bounce back-and-forth in the inlet tract, especially at part-throttle, and analysis of its performance is hampered by the chaotic nature of its movement: if the inlet valve closes before the pressure wave returns (because the piston speed is too low or the inlet tract too long) the net effect is reduced.

Suffice to say that anything that can be done to improve the momentum of the fuel/air charge during its passage from the carburettor to the cylinder will lead to an enhancement of torque; not only peak torque (although due to the low pressure involved there is a finite limit to the maximum level – around possibly 154–160lbf ft on the normally aspirated 8v 2*l*), but also the spread of torque.

The pressure differential across the valve throat sets the charge in motion – it rushes from high pressure (atmospheric) to the low-pressure cylinder. As the airstream passes through the choke its velocity increases and the pressure drop created causes fuel to be discharged from the main jet by atmospheric pressure acting on the float chamber. The response of the carburettor to this 'signal' is crucial to torque production. With a single-carb set-up, there will be conflicting pressure waves in a four-branch inlet manifold with competition cams, and interaction between the waves causes loss of top-end torque by comparison because the air momentum to the individual cylinder is degraded.

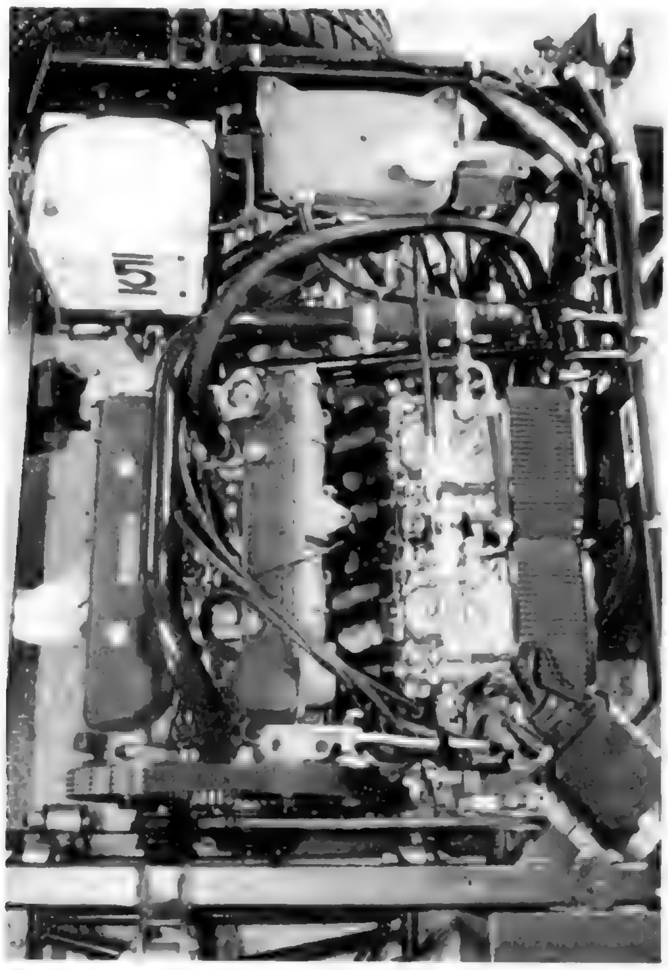


10/1: Split intake tracts reduces interference between cylinders.

When the inlet valve closes, it is advantageous if the air momentum in the tract is sustained to build up an 'over-pressure' behind the valve. Again, the one-choke-per-cylinder layout assists this process because air is not 'robbed' from one inlet tract to another. Matching of the inlet tract length and rampipe design to the specification of the TC as a whole yields very real and measurable results, and must be considered where all-out power is required. This momentum, also known as column inertia, is lost when short inlet branches are linked, as on the fuel-injected models, and it is no coincidence that manufacturers are experimenting increasingly with long inlet branches downstream of the common plenum and throttle body.

As with exhaust manifolds, no one inlet tract length (between the valve and carb mouth, or carb and rampipe mouth) will give optimum results throughout the whole rev-band. However, by careful selection, it is possible to achieve a length compatible with the changing frequency of the wave to ensure it arrives at the valve at the right time in the intake cycle at certain speeds in the rpm range.

A comparison of two GCT engines *(Case History Nos 2 and 7)* demonstrates this principle – two different cam/carb set-ups on otherwise almost identical engines, and two radically different inlet



10/2: GC Beta sidedraught manifold fits easily into frame of JH Classics (now Deon Cars) 246 Replica. On flowbench tests this manifold (matched to the head) only lost 1 cfm over 'straight-shot' manifold on flowbench tests. Manifold is offset to clear block-mounted distributor and has 20° downward tilt to cope with Beta installation. Carburettors are 45 DCOEs, K & N filters. Bosch ignition pack is visible at lower right.

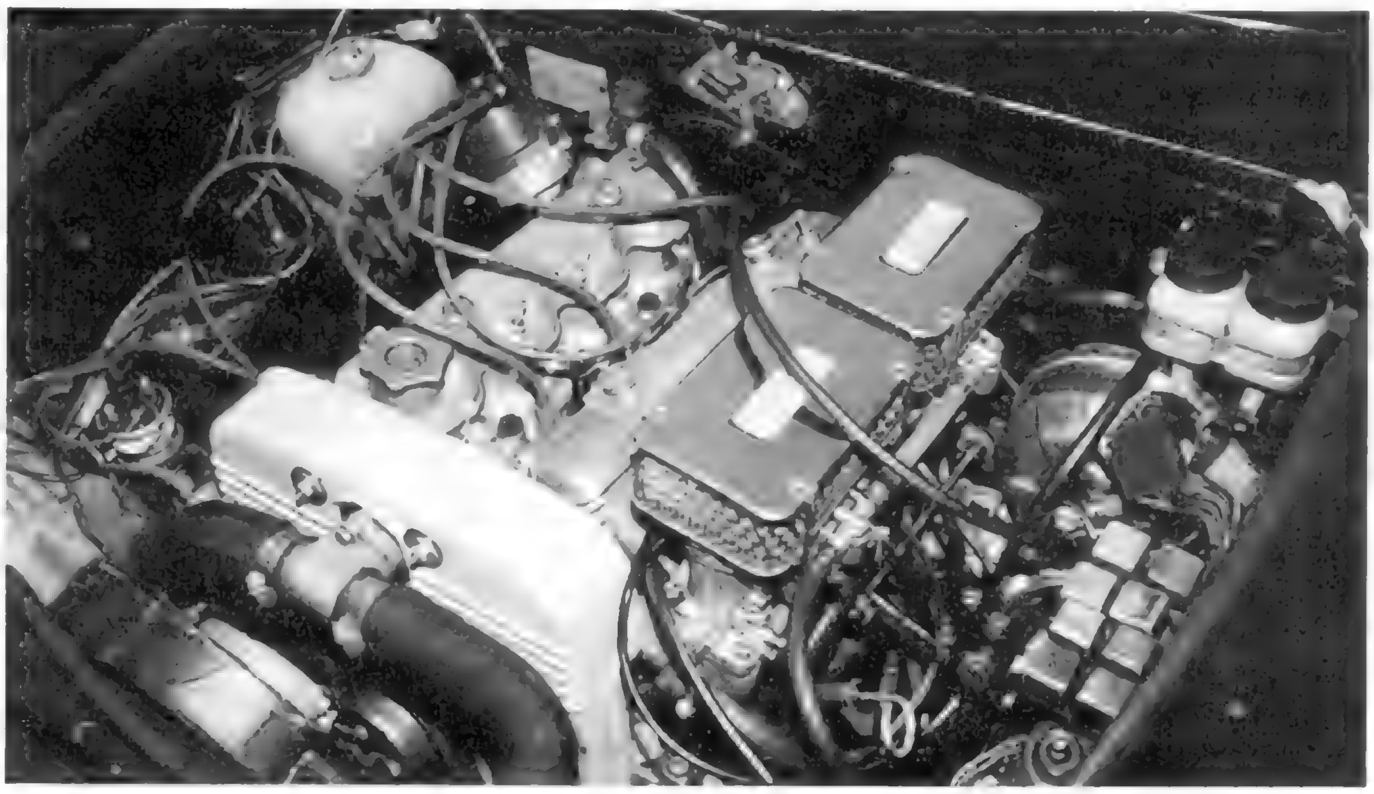
tract lengths (Thomas – 2¾" manifold, Casey – approx 8¾"), but peak torque results within 2lbf ft of each other. The major difference between the fuel systems was that the 48 DCOE used on Tom Casey's engine resulted in a lower air velocity and hence loss of column inertia when used with the original short manifold. Maintenance of air velocity is critical.

The size of carburettor for a given engine is heavily dependent not only on the valve size and cam configuration, but also the relationship of the inlet bore/length. GCT have always tended to use an included angle of not more than 14°, since beyond this angle the airflow breaks away from the port walls and becomes turbulent. **(10/6)**

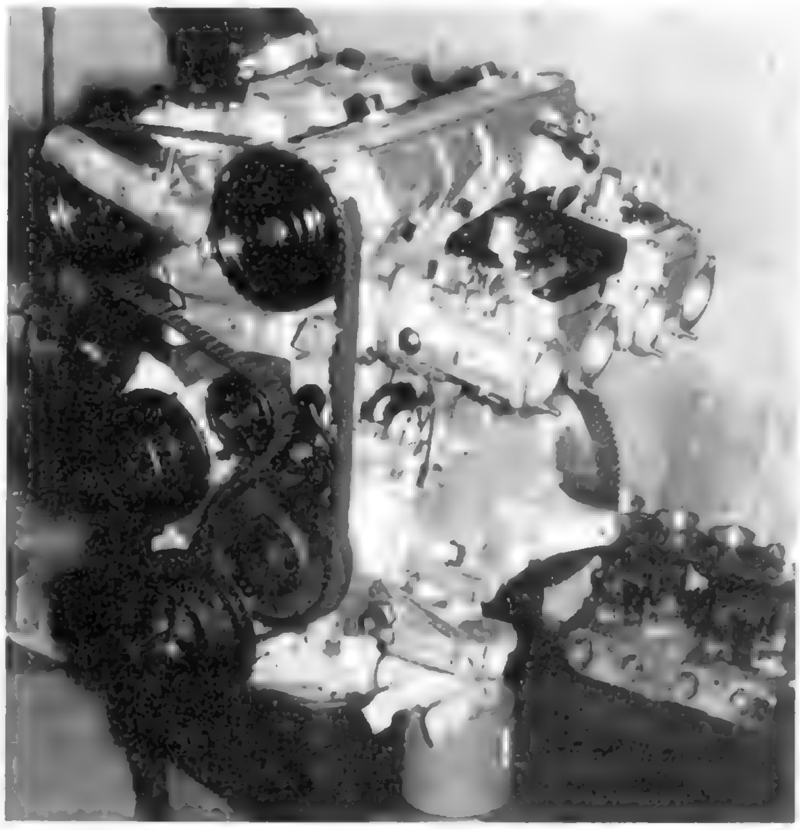
The port dimension is primarily dictated by the airflow (cfm) characteristic *(see Chapter 5 – Cylinder Head Preparation)* and therefore it follows that the carb size is going to be primarily dictated by this relationship. This tapered design tends to accelerate the air in the inlet tract and raise its speed and hence momentum. If the carburettors are too large, the airspeed through them will be too low and torque may suffer drastically.

The carburettor is required to produce an accurate fuel/air ratio at a range of

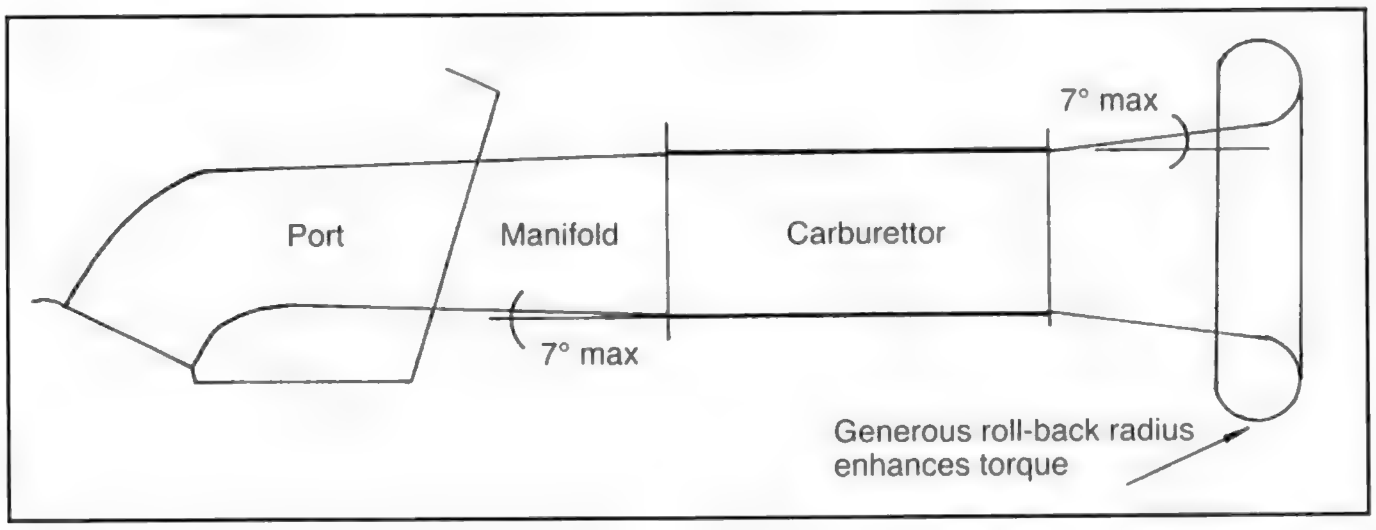
10/3: Simon Cole's nicely presented St II 2l Beta Coupe seen at Italian Intermarque Challenge features 44 DCNFs with deflector plate to keep warm air from radiator away from carbs. Regulator/filter is Filter King, from Malpassi.



10/4: Julian Sudano's US 124 Spider sports IDF carbs on production Fiat manifold with top-mounted distributor. IDFs function in same way as DCOEs with exception of pump jets, which are activated by a diaphragm unit.



10/5: 2l Monte Carlo engine at GCT in process of preparation for Duckhams/QXR car. Straight-shot Monte Carlo manifold, 48 DHLA. Note distributor swapped for exhaust cam-driven type – cams were purloined from author's own car! Matching pedal movement to throttle lever can be a problem on MCs. A small bell-crank assembly may be needed to amplify movement.



10/6: Port taper.

speeds, but inevitably part-throttle performance (especially in the progression phase) tends to be inferior in terms of economy because of the high pumping loss around the throttle plate and poor fuel signal. Correctly calibrated carburettors, however, do represent a triumph of ingenuity. At closed throttle, the inlet manifold is at a high state of vacuum, which drops progressively as the throttle is opened. The initial high rate of change when the throttle is 'snapped open' causes the fuel to condense on the walls of the manifold, starving the cylinder and tending to create a sudden torque loss ('flat spot'). For this reason, all the standard carbs (and competition types listed earlier) are fitted with pump jets, which spray neat fuel into the system to compensate.

Production models using a single carburettor utilize heated inlet manifolds to ensure that each cylinder receives equal proportions of the various fractions (with their different distillation boiling points) of petrol, since these fractions have differing octane ratings (a fuel of 96 octane RON may have fractions with boiling points of between 80°F and 385°F, with octane ratings from 91–97).

There is no such problem with twin-carb set-ups (although in cold climatic conditions manifold icing may occur and warm-air ducting from the exhaust may be needed) since the fuel distribution is incomparably superior, but the curious effects in the standard inlet manifold can (and frequently do) lead to detonation problems in No 4 cylinder on production, single-carburetted engines. This heating leads to a loss of air density (and power) and competition types should have the coolant galleries blanked-off and the engine jetted deliberately rich.

With twin carbs it is perfectly satisfactory to use 40s on a *highly tuned* 2*l*: a good spread of modest torque will result, although the airflow restriction from the small carbs and limited air momentum will weaken the top-end power compared with, for example, 45s or 48s. Using a single twin-choke carb on a highly tuned engine, however, can cause the airflow to go supersonic in the small secondary chokes. When this happens, instead of a pressure drop being created, the choke becomes pressurized and the fuel cannot pass into the airstream; a larger carburettor or specially located 'power enrichment' jets will be needed *(see Case History No 4)*.

Conversely, the progression from idle to main fuel circuits tends to be smoother with a single carburettor (in mild states of tune – *eg* standard, St II cams) because of the high airspeed generated in the venturis by the combined pumping action of the four cylinders. Therefore, if the special tuning requirement is for a very flat low-down torque curve, a single carb may be the answer.

Race carburettor options

GCT tests indicate that, because of the adverse reflections generated in downdraught manifolds owing to the 90° bend in the tract, sidedraughts will generally yield about 3–5% extra torque *per se*, though, that said, there are installations where sidedraughts cannot be fitted because of the absence of sufficient space. **(10/7)**

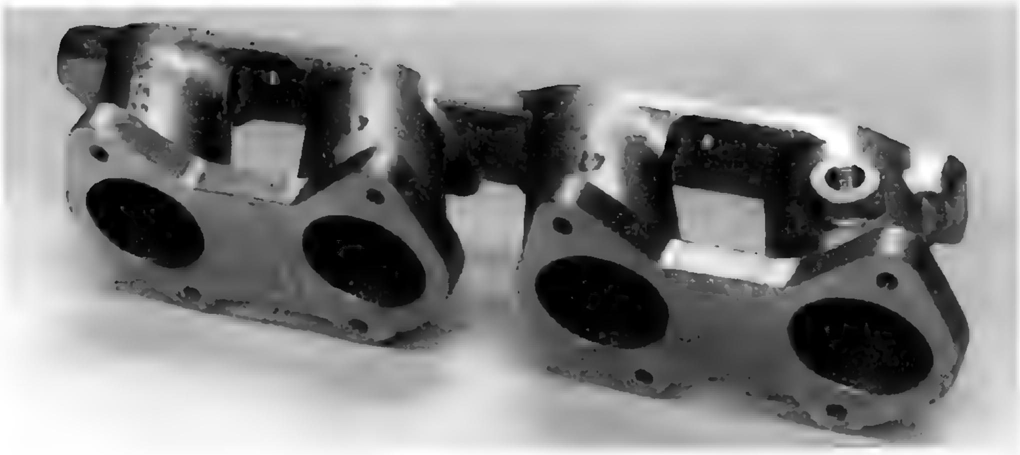
Always allow at least 1½" from the carburettor mouth to the nearest bulkhead or chassis intrusion to allow for the fitment of (short) rampipes and shallow filters. If the initial negative wave is reflected off a solid air filter casing located too close to the carb mouth, torque will be reduced as it will be reflected as a negative, rather than a positive wave.

IDFs have been used (on a works X1/9 development car) on transverse installations, but their float chambers are in fact designed for in-line engines. DCNFs are considered to be the best choice for this layout. DCOE and DHLA are totally interchangeable for sidedraught applications, as are IDF and DRLA for downdraught (RWD). Sidedraughts must be mounted on special fitting kits to prevent engine vibration, which can cause the fuel to froth in the float chamber. Downdraughts are bolted direct to the manifold, but should ideally be mounted on phenolic insulator blocks to minimize heat transfer.

Sidedraughts must be fitted within 5° of horizontal and downdraughts should be mounted vertically. (GCT have used a 7° angle on sidedraughts satisfactorily, *eg* on Westfields, where the chassis intrudes.)

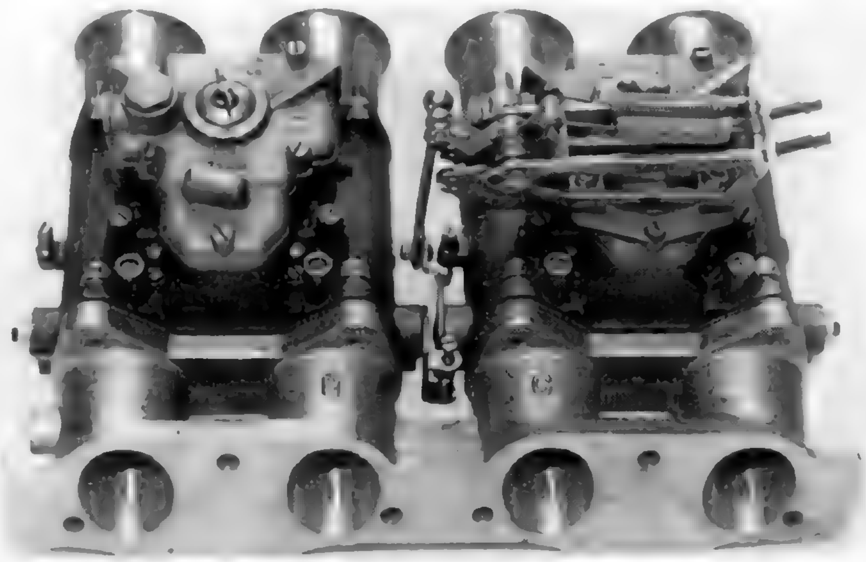
10/7: If you have a LHD 124 with block-mounted distributor, you need an offset, downdraught manifold for a Weber IDF (shown) or Dellorto DRLA. Items shown came from Italy. IDFs are prone to dirt problems with air correctors exposed as shown – complete, all-enclosing air filter would be better; additionally, flow loss through gauze is high, filtration value nil! Conversion of LHD 124 to sidedraughts requires brake cylinders to be moved into scuttle, 'works-style', and servo removed.

10/8: GC 131 sidedraught manifold. Fits all 8v TC models, especially those with block-mounted distributor. 2¾" long, LM4 pure ingot casting – strong, but easy to open out if required. This manifold is cast with shaped cores and is virtually a perfect match for heads (although port sizes vary). Servo take-off accepts standard Fiat/Lancia fitting M14 x 1.5. Although designed for 45s, this manifold works perfectly satisfactorily with 40s.

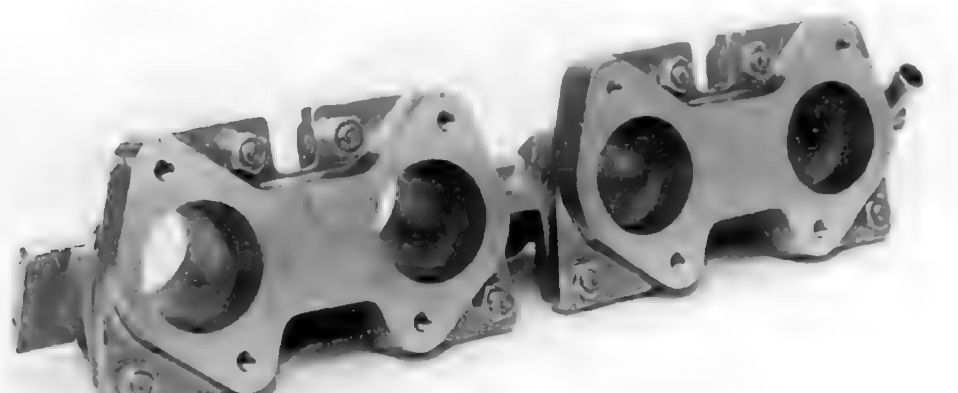


10/9: GC Beta manifold for 2l engine. Due to proximity of breather and starter motor, 1600 model requires 131 type with 'swan neck' wedge plates fitted to outer face; 2¾" long.

10/10: GC Monte Carlo sidedraught manifold. Only 2¼" long with top-mounted distributor, no offset required. Gave only 1 cfm extra airflow over offset design; main advantage is smaller size. Can be machined to fit upright 131-type engine (with top-mounted distributor) – 1¾" in straight-shot layout. This straight-shot version can also be used on Strada 130 TC.

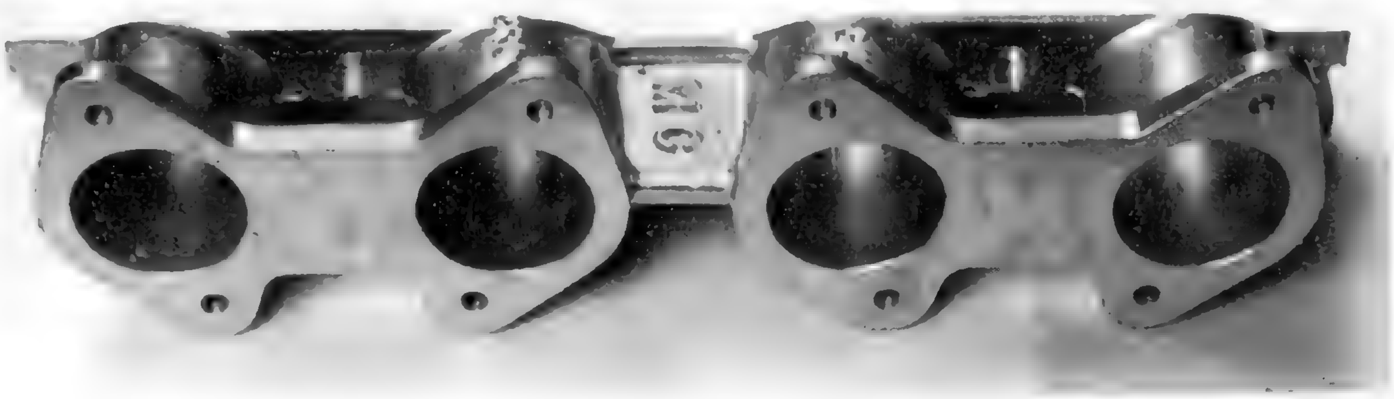


10/11: Monte Carlo manifold opened out to full-race and fitted with 48 DCOE. Much 'fiddling' required to get throttle linkage to work, despite use of versatile (and well-designed) Weber type 48s. DCOEs require special nuts to bolt to manifold – 12mm (across flat) because of large throttle body. 'K' nuts from aerospace supplier are ideal. Rampipes are standard DCOE types (see **10/51**).

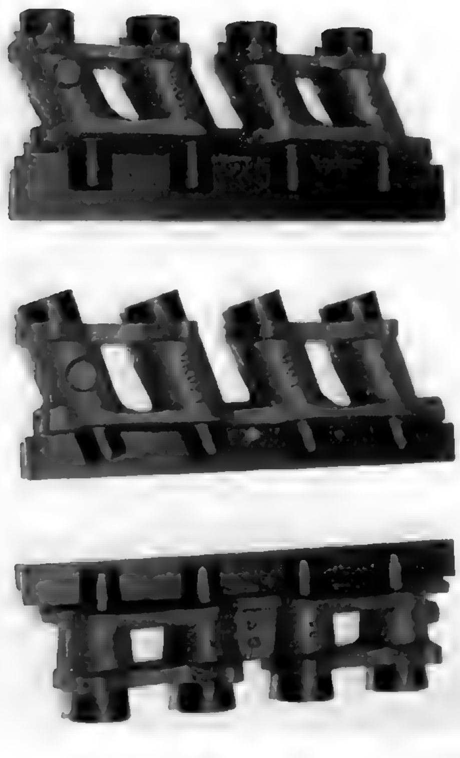


10/12: GC 130 TC adaptor plates allow 45s to be bolted straight on to engine. Diecast baseplate is retained. Plates also work perfectly with 40s despite 2½mm mismatch on radius. Plates are interchangeable with OE items – many of which are now suffering from perished rubbers.

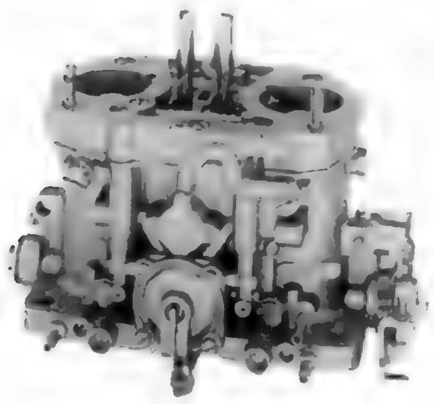
10/13: Optimum fitting kit: Misab bonded nitrile rubber O-rings, nyloc nuts, studs, Cosworth isolator washers.



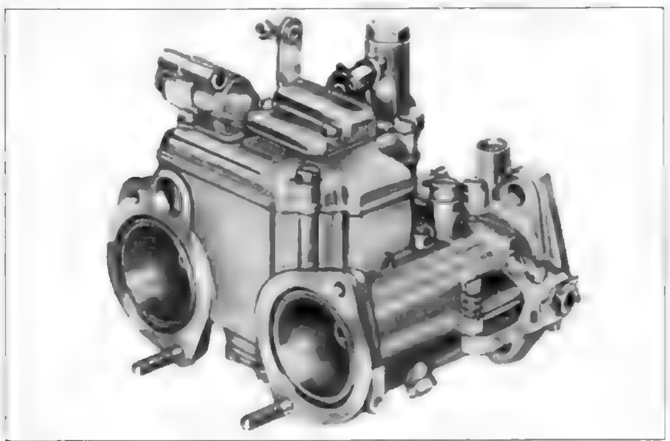
10/14: Straight-shot 131 manifold, 1¾" long.



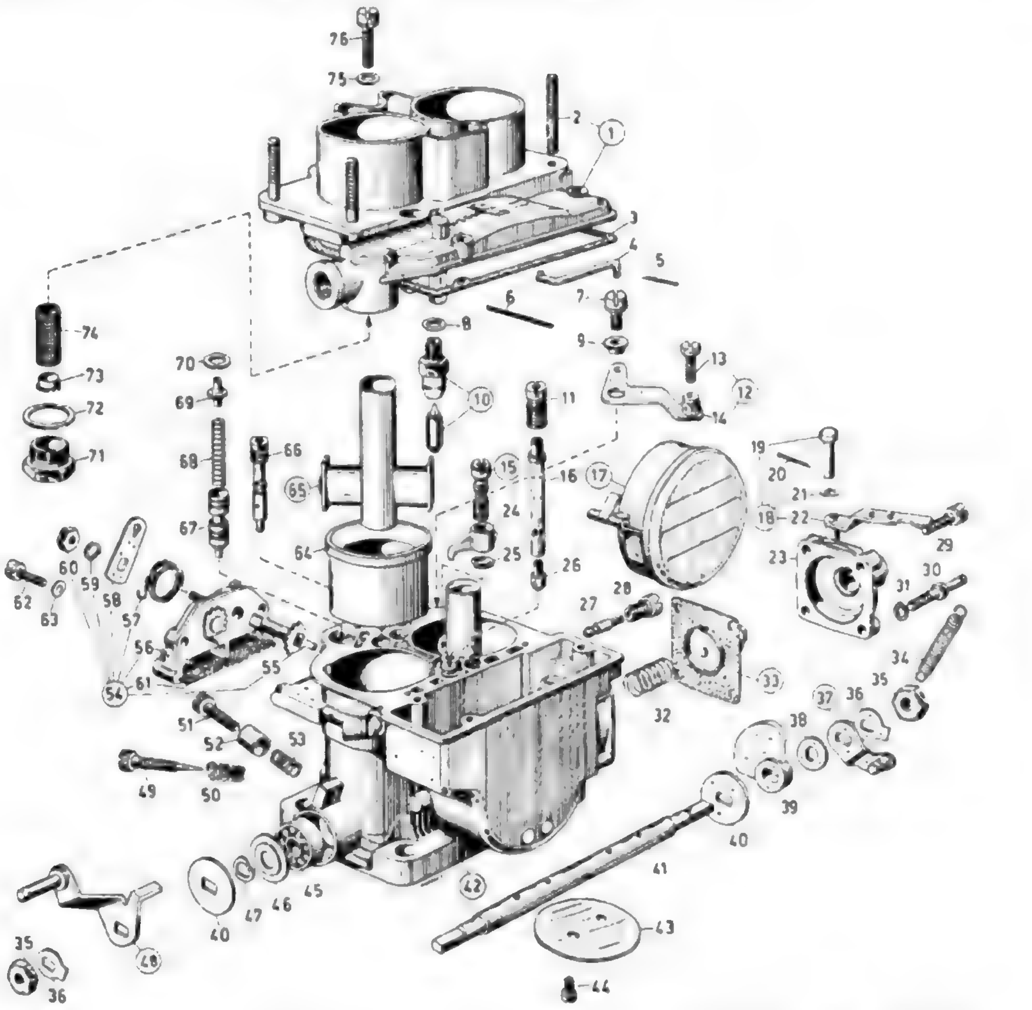
10/15: Top-quality hardwood patterns made by Brian Bonner of A C Spinks (Gravesend) will permit thousands of manifolds to be made if they are well looked after. Patterns are split down middle; each half is placed in sand-filled casting box and sand cores fitted in place to produce manifold bores. Boxes are fitted together and LM4 casting aluminium poured in.



10/16: Dellorto DRLA.



10/17: Dellorto DHLA.

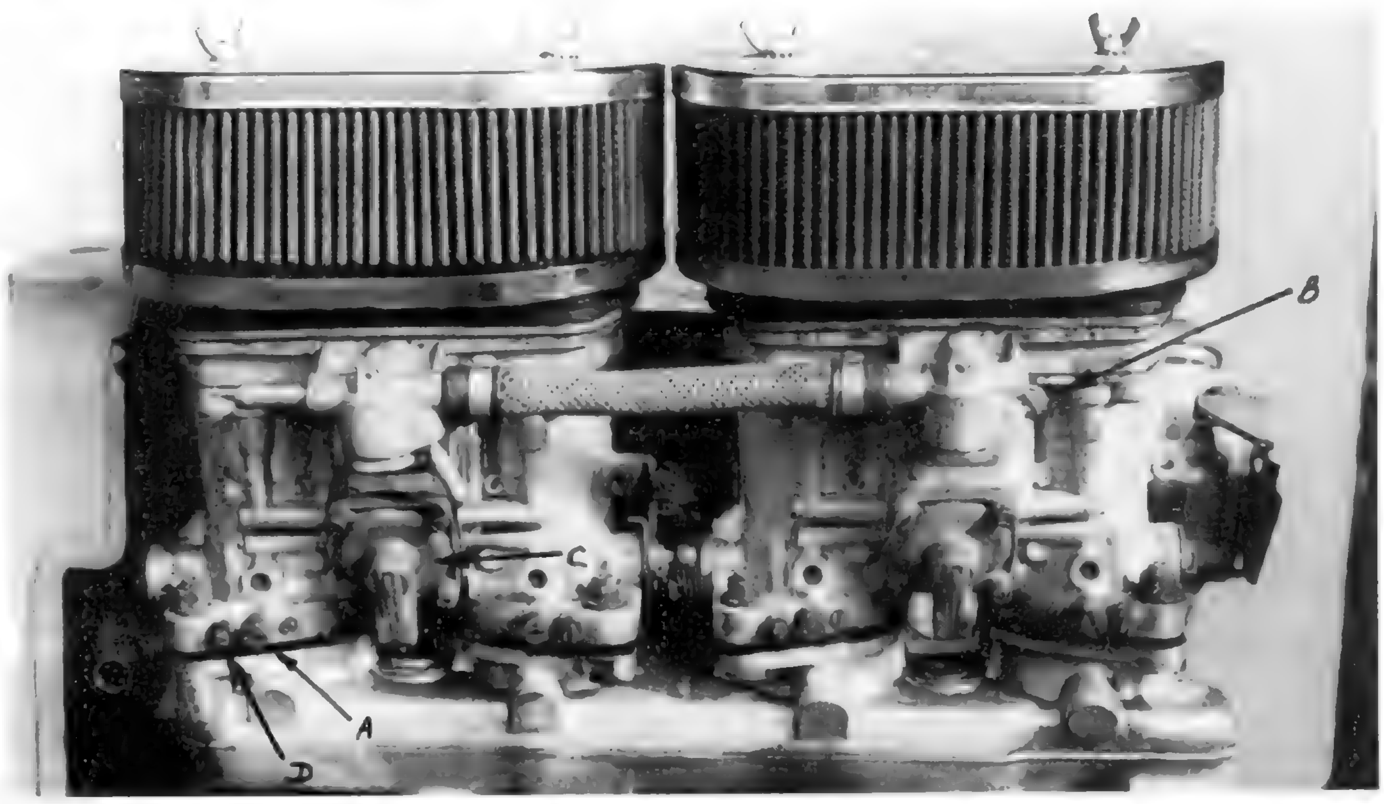


1 Float-bowl cover
2 Stud bolt
3 Gasket
4 Choke rod
5 Cotter pin
6 Float-fulcrum pin
7 Bolt lever
8 Needle-valve gasket
9 Bushing lever
10 Float needle valve
11 Air-correction jet
12 Lever assy.
13 Choke-cable lock screw
14 Lever
15 Accelerator-pump assy
16 Emulsion tube
17 Float
18 Accelerator-pump-cover assy.
19 Pin
20 Cotter Pin
21 Wave washer
22 Lever-accelerator pump
23 Accelerator-pump cover
24 Accelerator-pump jet
25 Accelerator-pump gasket
26 Main jet
27 Idle jet
28 Idle-jet holder
29 Accelerator-pump-cover bolt
30 Spring mount
31 Lock washer
32 Pump loading spring
33 Accelerator-pump-diagram assy.
34 Throttle return spring
35 Throttle-shaft nut
36 Washer
37 Throttle
38 Washer
39 Accelerator-pump cam
40 Washer
41 Throttle shaft
42 Carburetor body
43 Throttle valve
44 Throttle-valve screw
45 Throttle-shaft bearing
46 Cushioning plate
47 Wave washer
48 Throttle-valve lever
49 Idle-mixture screw
50 Spring
51 Idle-speed screw
52 Spacer
53 Spring
54 Choke-cover assy.
55 Choke drive gear
56 Choke cover
57 Choke return spring
58 Choke lever
59 Lock washer
60 Nut
61 Filter element
62 Bolt
63 Flat washer
64 Venturi
65 Auxiliary venturi
66 Starter (choke) jet
67 Choke valve
68 Choke-valve spring
69 Spring guide
70 Lock washer
71 Filter cap screw
72 Gasket
73 Filter cap
74 Filter
75 Washer
76 Screw

10/18: Components of Weber DCNF.

Layouts (10/16 – 10/19)

In-line engine – rear-wheel drive
Sidedraught:
Weber DCOE – 40, 45, 48
Dellorto DHLA – 40, 45, 48
Solex ADDHE – 40
Also feasible (but not yet tried by GCT): SU and Mikuni. (The SU variable-choke carb is ideally suited to supercharged set-ups, but calibration is more complicated than the DCOE.)
Downdraught:
Dellorto DRLA – 40/45
Weber IDF – 40, 44, 48
Weber IDA – 40, 48
Transverse engine
Sidedraught:
As above
Downdraught:
Weber DCNF – 40/44



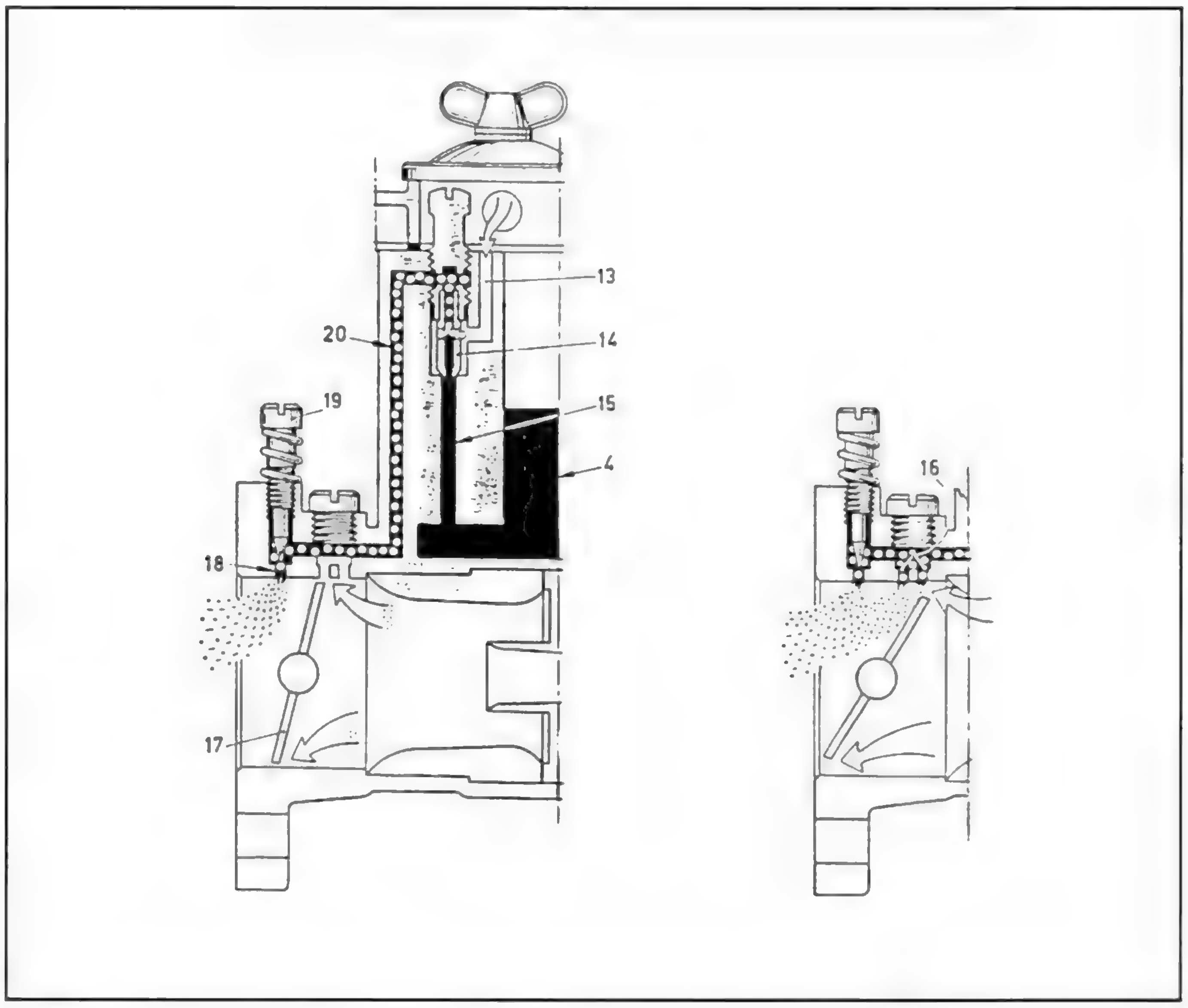
10/19: 40 IDF on 1608 124 BC, photographed at GCT. Arrowed: A – idle mixture screw; B – idle jet; C – pump jet diaphragm unit; D – balance screw (one per choke).

Weber DCOE mode of operation (10/20 – 10/25)

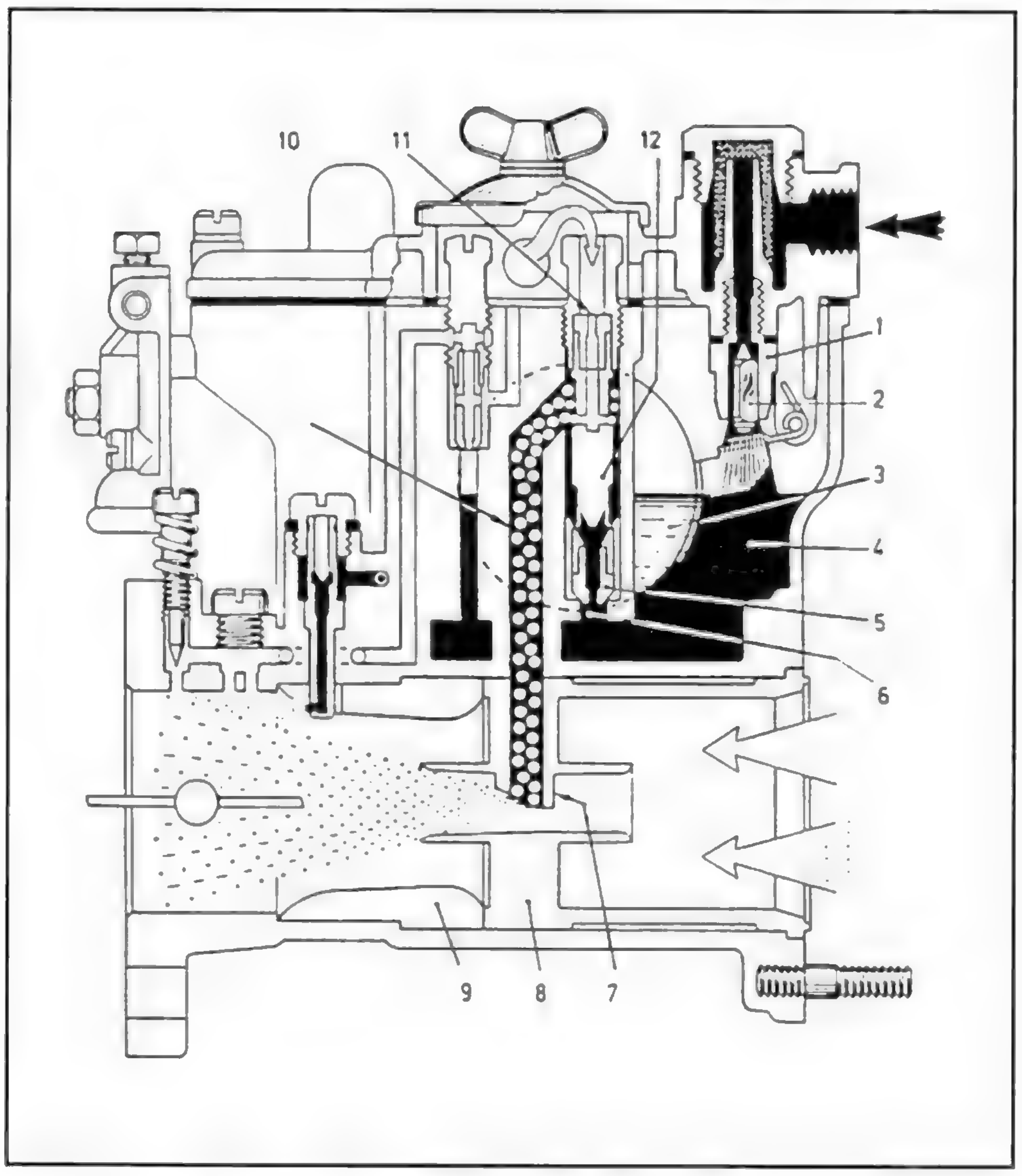
The layout of the world-famous Weber DCOE serves well to illustrate the general mode of operation of the various types of race carburettor, albeit that there are design differences between models.

Float levels

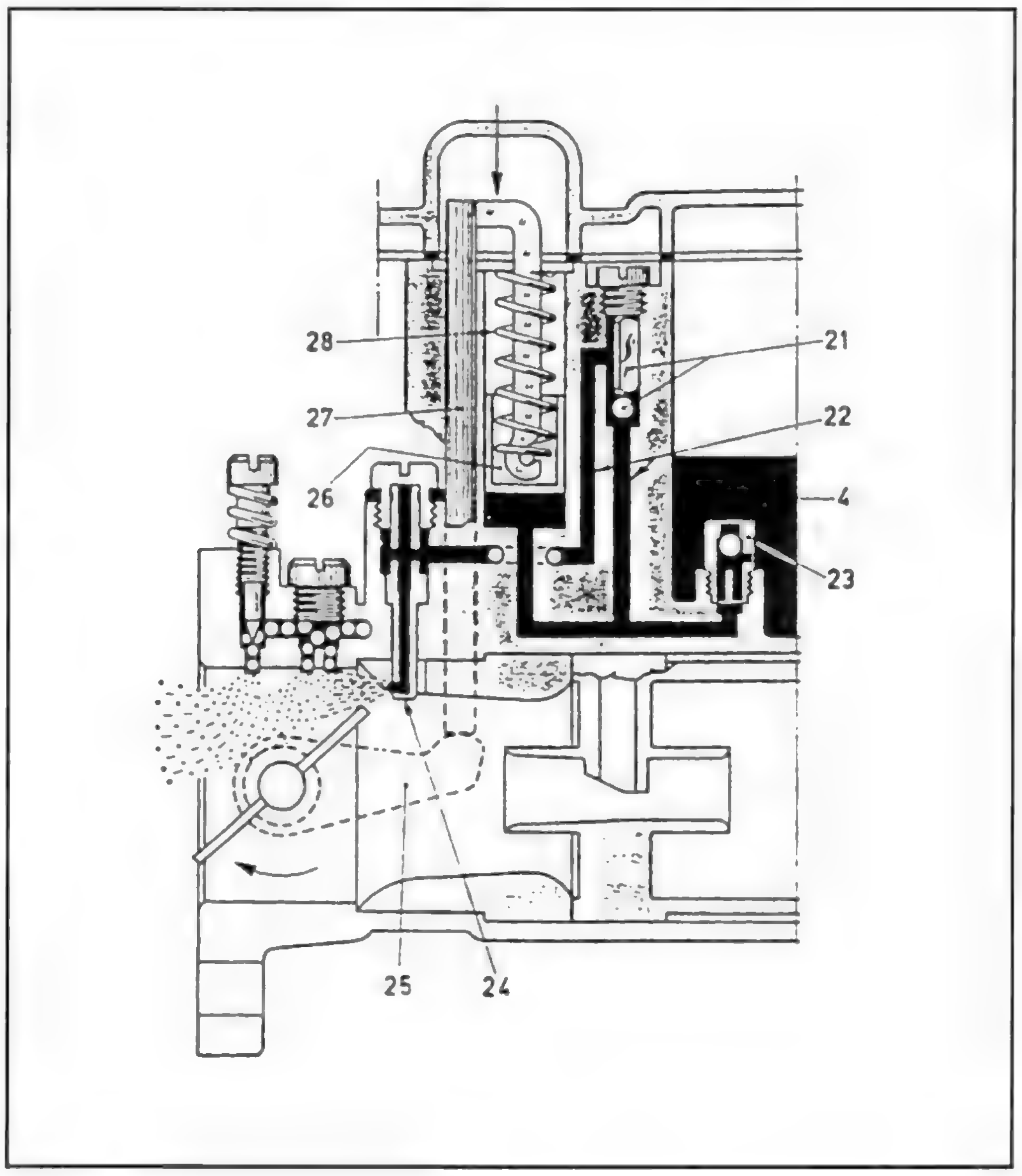
The fuel level in the float chamber is critical: too low and the main/idle emulsion tubes will receive insufficient fuel; too high and flooding will result. Both conditions lead to power loss, jetting problems and very likely engine damage (especially flooding). If the float levels are upset because the carbs are incorrectly mounted, the same results can be expected.



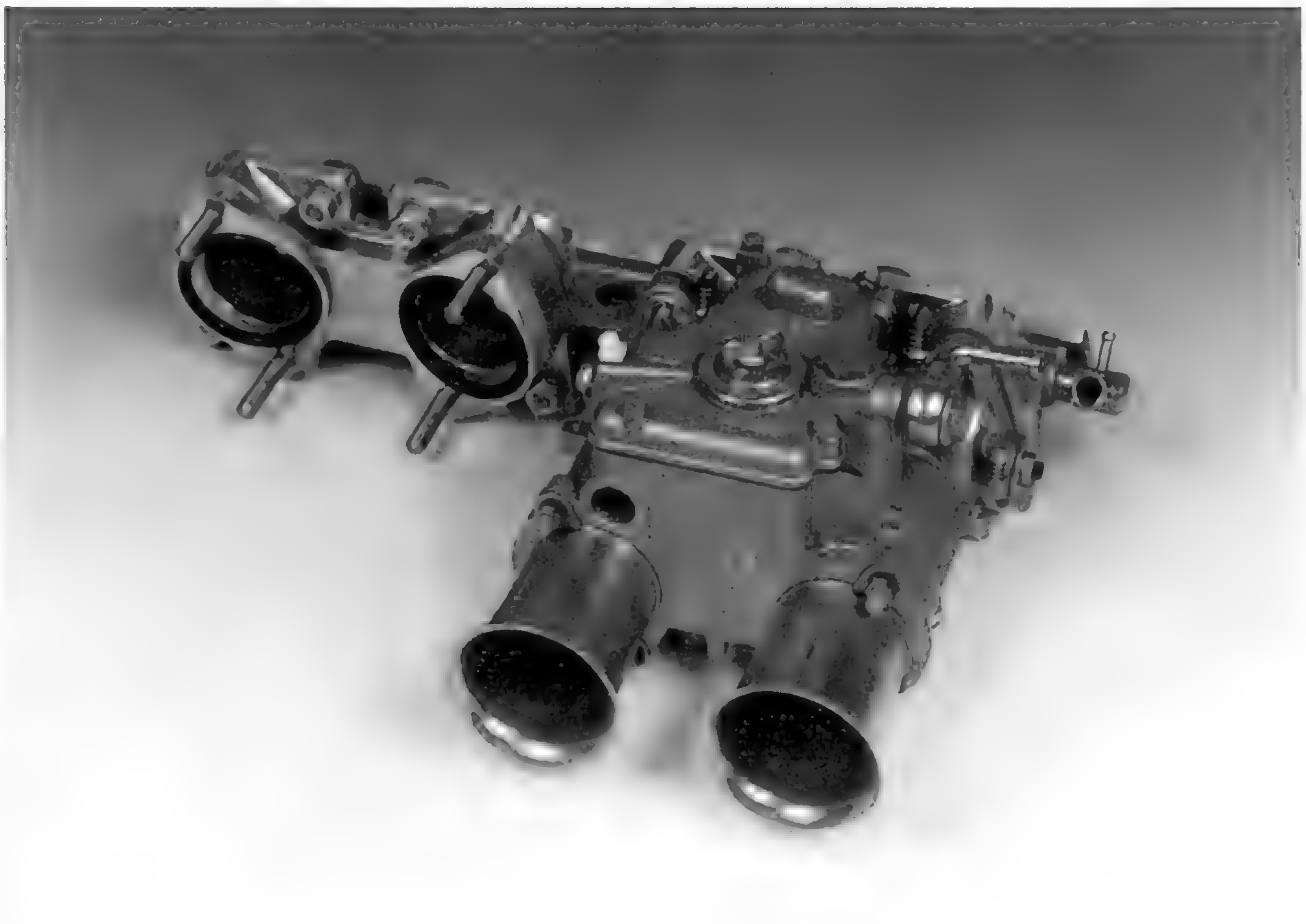
10/20: DCOE IDLING OPERATION AND PROGRESSIVE ACTION
Fuel is carried from bowl (4) to calibrated holes and idling jets (14) through ducts (15). Emulsified with air coming from ducts (13), through ducts (20) and idling feed holes (18), adjustable by means of screws (19), fuel reaches carburettor throats below throttles (17). From ducts (20) mixture can also reach carburettor throats through progression holes (16).



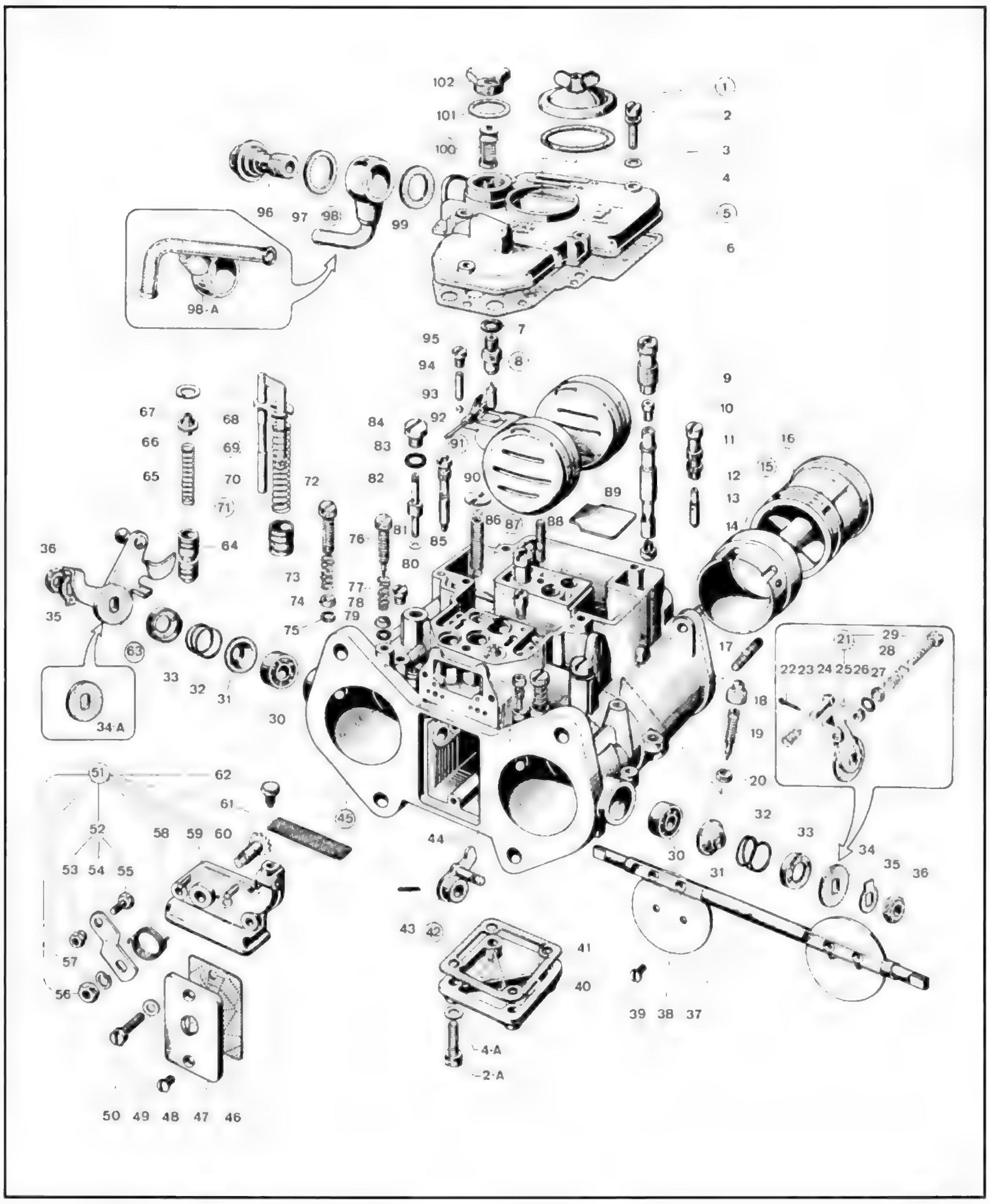
10/21: DCOE NORMAL OPERATION
Fuel arrives through needle valve (1) to bowl (4), where float (3) controls opening of needle (2) in order to maintain constant fuel level. Through ducts (6) and main jets (5), it reaches emulsioning tubes (12), from which, after being mixed with air from air corrector jets (11), through pipes (10) and nozzles (7), it reaches carburation area consisting of auxiliary venturis (8) and chokes (9).



10/22: DCOE ACCELERATION
By closing throttle valves, lever (25), by means of shaft (27), lifts piston (26). Thus fuel is drawn from bowl (4) into pump cylinder through suction valve (23). By opening throttles, shaft (27) is free and piston (26) is pushed down under action of spring (28); by means of ducts (22), fuel is injected through delivery valve (21) to pump jets (24) and into carburettor throats. Inlet valve (23) is provided with a calibrated hole which discharges excess fuel delivered by accelerating pump into float bowl.



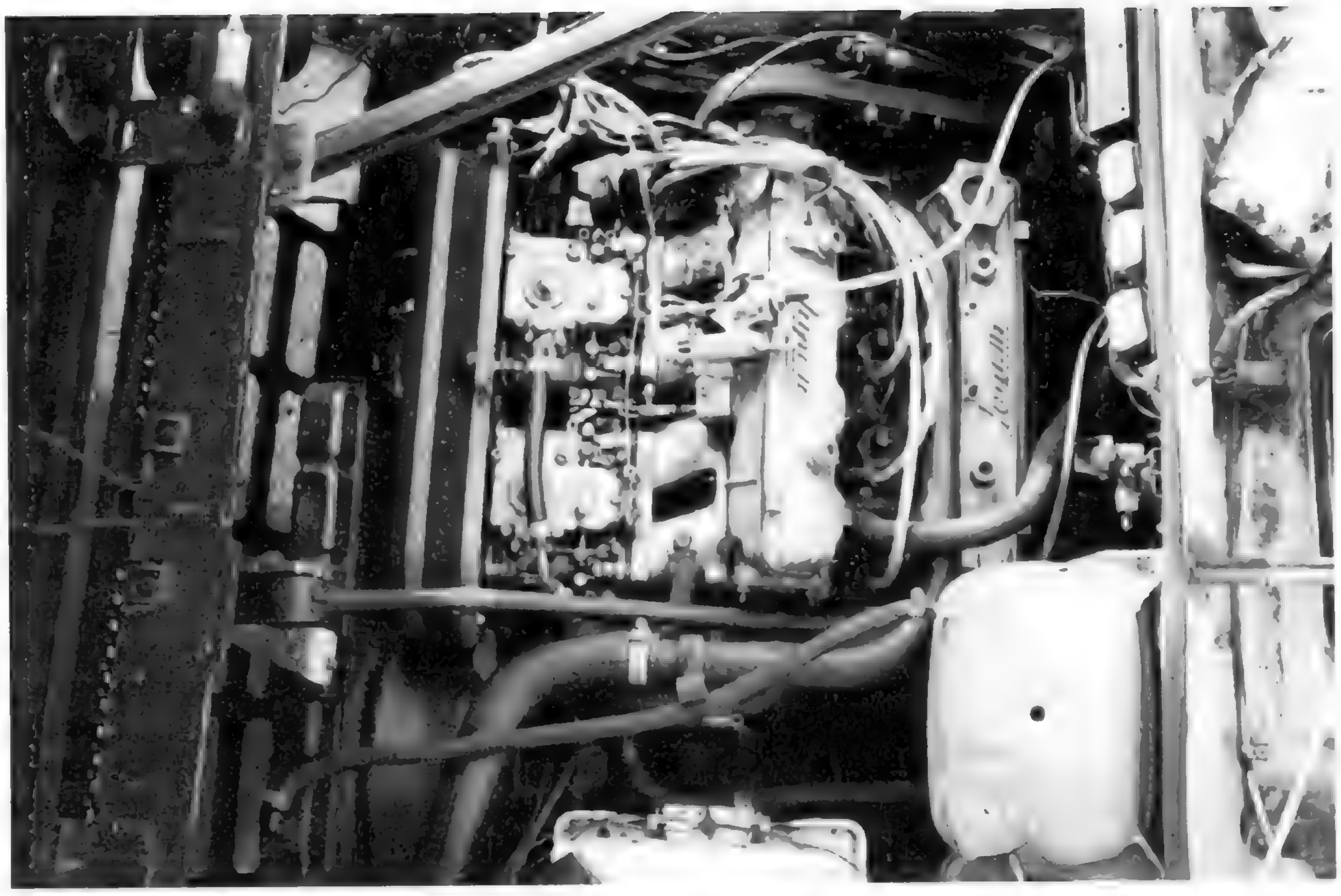
10/23: 130 TC baseplate manifold fitted with GC adaptor plates and 45 DCOE. Unlike hard to get DHLA Dellorto carbs (now out of production), Webers do not come as a pair. Throttle spindle balance quadrant needs to be fitted to RH carb and linkage quadrant to LH carb. A fuel T-piece is also required. Plastic cap just visible on left side of carb covers air bypass screws (for balancing chokes relative to each other).



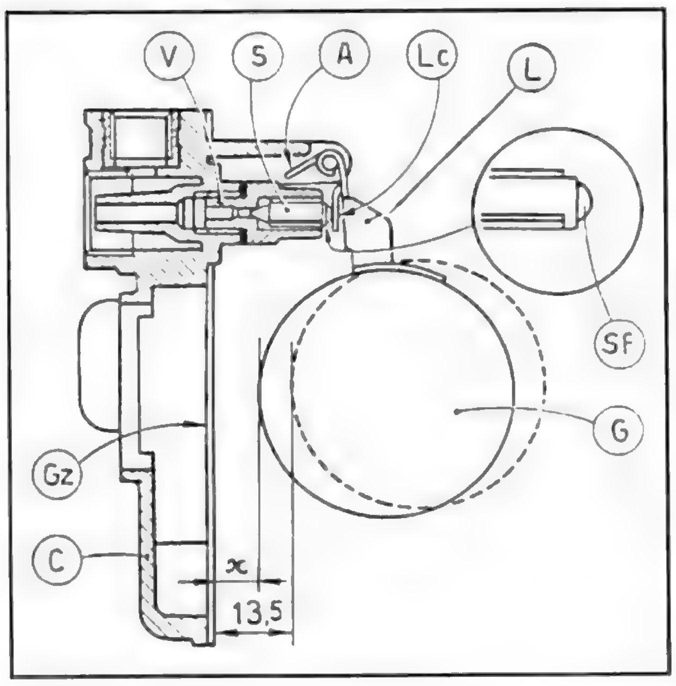
10/24: WEBER DCOE EXPLODED VIEW – key to main components

1 Jet inspection cover
2 Top cover screw (5)
3 Gasket
4 Spring washer (5)
5 Top cover
6 Gasket
7 Sealing washer
8 Needle valve
9 Main jet holder (2)
10 Air corrector (2)
11 Idle jet holder (2)
12 Emulsion tube (2)
13 Idle jet (2)
14 Main jet (2)
15 Primary choke (2)
16 Secondary choke (2)
17 Stud (4)
18 Air balance screw cover (2)
19/20 Air balance screw and nut
21 Balance quadrant assembly
30–36 Bearing assembly (2)
37 Throttle spindle
38/39 Throttle plate (2) and screws (4)
40/41 Bottom cover and gasket
42/43 Pump actuating lever
44 Pump jet
45 Carb body
46–50 Cover assembly
51 Starter valve actuating assembly
63 Throttle lever
64–67 Starter valve assembly (2)
68–70 Pump jet actuating rod assembly
71 Pump jet piston
72–75 Idle speed screw assembly
76–79 Idle mixture screw assembly (2)
80 Progression hole cover (2)
81–84 Pump jet assembly
85 Pump jet piston retaining plate
86 Throttle return spring
87 Pump jet bleed valve
88 Stud
89 Vent cover
90 Starter jet (2)
91–92 Float and fulcrum pin
93–95 Pump jet non-return valve assembly (2)
96–99 Banjo union assembly
(98A – T-piece)
100–102 Carb filter assembly

10/25: Will 45 DCOEs fit Lancia Beta? Engine bay of Marious Lourides' car shows how! Choke cable is redundant really – three or four full pumps of throttle are usually all that is needed. Linkage is adapted from Beta item. A cold-air box with remote (turbo-type) filter would be better because of proximity of radiator.



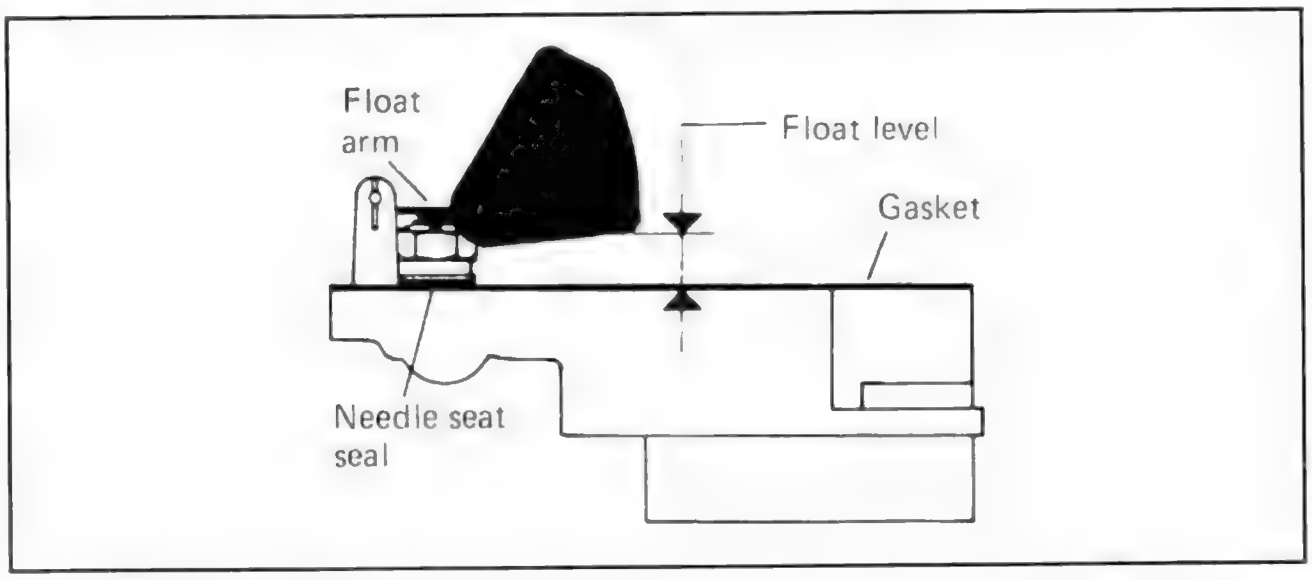
DCOE float level (10/26)



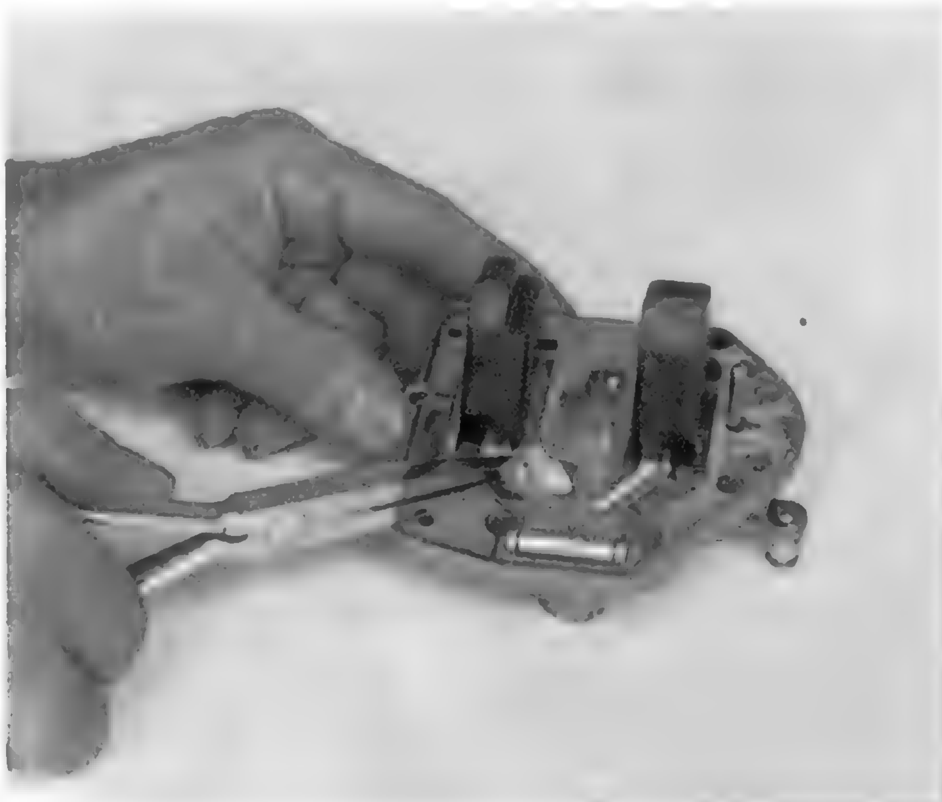
10/26: DCOE SETTING FLOAT LEVEL
Hold carb cover near vertical so float (G) is in light contact with needle valve (Sf). Gasket (Gz) is in place. Measure distance between both floats and gasket (x) with float arm in light contact with ball of needle valve. Settings vary on carbs from donor vehicle (eg Alfa Romeo), but for current production aftermarket carbs, use following:

	45	40/48
Brass float	5mm	8mm
Black plastic floats	12mm	12mm

Solex 40 (130TC) float level (10/27 – 10/29)



10/27: Diagram of layout of components determining float level. (Fiat Auto SpA – copyright reserved)



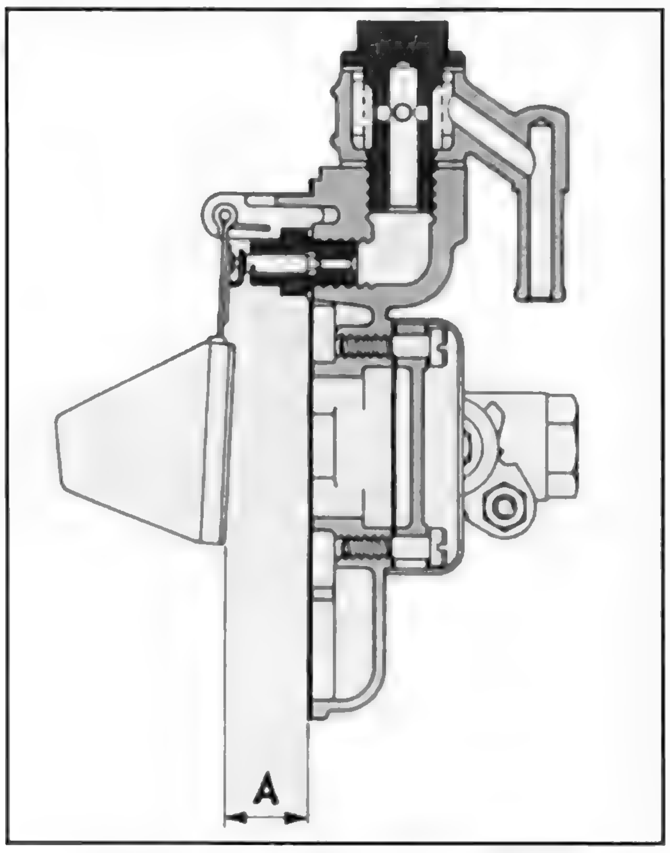
10/28: Float level is checked with carb cover in horizontal position with weight of float completely depressing needle valve ball. Distance between float and plane of cover with gasket fitted should be 3.5–4.5mm. (Fiat Auto SpA – copyright reserved)

10/29: If level does not correspond to value given it is necessary to alter thickness of seal under needle seat or adjust float arm. (Fiat Auto SpA – copyright reserved)

DHLA float levels (10/30)

The following data applies to all 40, 45, 48 DHLAs:

Float No	A
7298.1	14.5–15mm
7298.2	16.5–17mm



10/30: DHLA float level. Check that the float has the same weight as it has marked on it, is undamaged and is also completely free to swivel on its pivot pin. Hold the carburettor cover vertically so that the float arm is in light contact with the needle but with the spring in the needle remaining uncompressed. In this position, check that both the half-floats are the correct distance from the chamber cover measured to the top-cover gasket.

DRLA float levels

Measure as DHLA, distance 'A' should be 5–6mm.

IDF float levels

Measure as DHLA; the distance between the point on the top of the float furthest away from the cover (without gasket) should be 10mm. In the fully open position, this should be 32.5mm.

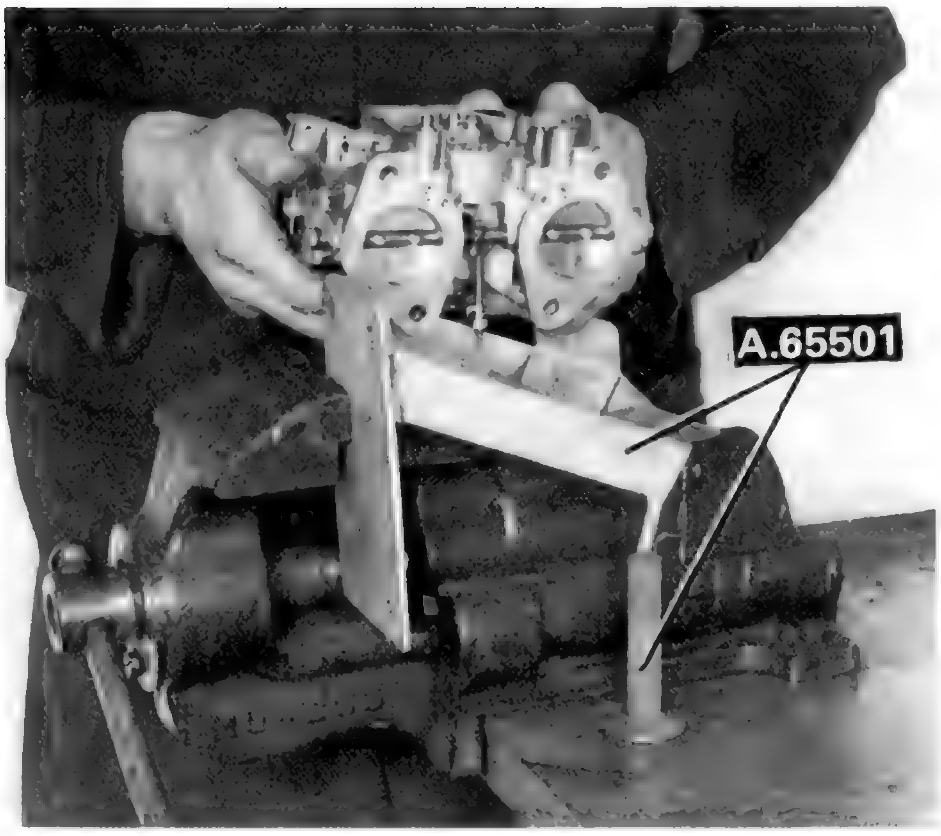
DCNF float levels

Hold the cover vertically, as with DHLA, so that the float arm is in light contact with the ball of the needle valve, and measure the distance from the cover to the bottom of the float. For 40, 44 DCNF this should be 50mm.

Allow the float to open fully; the distance should be 58.5mm.

Adjusting float levels – all types

To adjust the closed height, bend the float arm. To adjust the open height, adjust the tab which bears against the needle valve housing. If the correct height cannot be obtained, check that the sealing washer under the float needle housing is correct.



10/31: Checking flow rate of Solex 40 carb from Abarth 130 TC. After filling chamber with fuel main butterfly lever must be operated several times from min to max until circuit is completely full and there is a regular supply to pump injector. (Fiat Auto SpA – copyright reserved)

Pump jets (10/31)

Failure of the pump jet circuit to enrich the mixture during acceleration can lead to a serious flat-spot. On models fitted with diaphragm pumps (IDF, DHLA, DRLA, DCNF), ensure that the diaphragms are in good condition (if in doubt, replace). On DCOEs, ensure that the pump jet pistons move freely. If the delivery is inadequate, in addition to these checks ensure that the fuel galleries feeding the pump and pump jet are unobstructed. DCOEs in particular have a pair of small ball-weight assemblies between the pump piston and the jet – take care not to lose them if stripping (on refitting, the ball is fitted first).

New carbs obviously come pre-set, but if it is felt during tuning that extra enrichment is required during acceleration, there is provision for alteration. Likewise, when using 'used' carbs, this vital circuit should be checked before fitting to the engine.

The maximum delivery is controlled by the diaphragm unit (or piston, DCOE only). The stroke of the actuating arm can be adjusted for diaphragm types if required, and most models have a bleed-back valve in the float chamber which allows fuel to bypass the pump jet – thus reducing the size of this bypass valve will increase delivery, and similarly the pump jet can be changed (the largest GCT have had to use is 55 on a 48 DCOE – full-race).

Only Dellorto (and Solex – *see photo*) quote delivery amounts (on production carbs) – for Dellortos (40, 45, 48), 20 full open/close strokes of the throttle should produce 8 ± 0.5cc per barrel. This is easy enough to check by filling the float chamber and pumping the fuel into calibrated containers.

RACE CARBURETTOR JETTING

Requirements

Operations required when twin carburettors have been fitted are as follows:

1. Selection of choke size to suit engine specification
2. Selection of idle jet emulsion tube and main jet plus air corrector
3. Selection of needle valve and float type
4. Selection of pump jet
5. Setting of idle mixture and confirmatory checks on main jet/air corrector
6. Balancing of carburettors

Mixture levels

Fiat/Lancia Twin-Cam engines fitted with twin carburettors are very sensitive to mixture changes and, as a general rule, the mixture levels should be in the order of:

800–1000rpm: (no load)
2–3% (carbon monoxide)
400–800 ppm HC (parts per million hydrocarbon)

3000–max rpm: (full-throttle – full-load)
4½–5½% CO
150 ppm HC or less

CO

This is best measured using an infra-red carbon monoxide meter. CO is an inevitable by-product of the combustion process and can only be removed by use of a catalytic converter, which changes it to CO_2 (carbon dioxide). The level of CO in the exhaust is an accurate method of assessing the mixture level needed for a particular engine.

HC

Unburned fuel released into the tailpipe will show up on a hydrocarbon meter and will enhance the data from the CO meter. If the mixture is too lean the CO level will be low (because the mixture is too lean to burn), but the HC will be high, and *vice-versa*. Faulty plugs or ignition misfires, bent valves, etc, can lead to a high HC level even though the jetting level may be correct.

Importance of correct mixture

The Twin-Cam will develop maximum power fairly close to (but slightly richer than) stoichiometric (or chemically correct, *ie* 14.7:1) air/fuel ratio. A mixture which is too lean will reduce power and tend to cause overheating, pre-ignition and detonation. An over-rich mixture will cause valve seat damage (due to impact of carbon on valve seats), carbon deposits on combustion chamber, pistons and rings, and 'bore washing', where excessive fuel in the upper cylinders washes away lubrication, thus causing grossly exaggerated ring wear and piston scuff. Correct mixture will always correspond with maximum torque and hence bhp. Actual air/fuel ratios on TCs may vary from 12.5:1 at low revs (around 3000rpm) to 14.2:1 at high speed. The engine must be set up using the exhaust manifold and air cleaner for the final specification. An exhaust used for testing which is of the wrong type (*eg* 4-2-1 instead of 4-1) or too small will upset the mixture/power result.

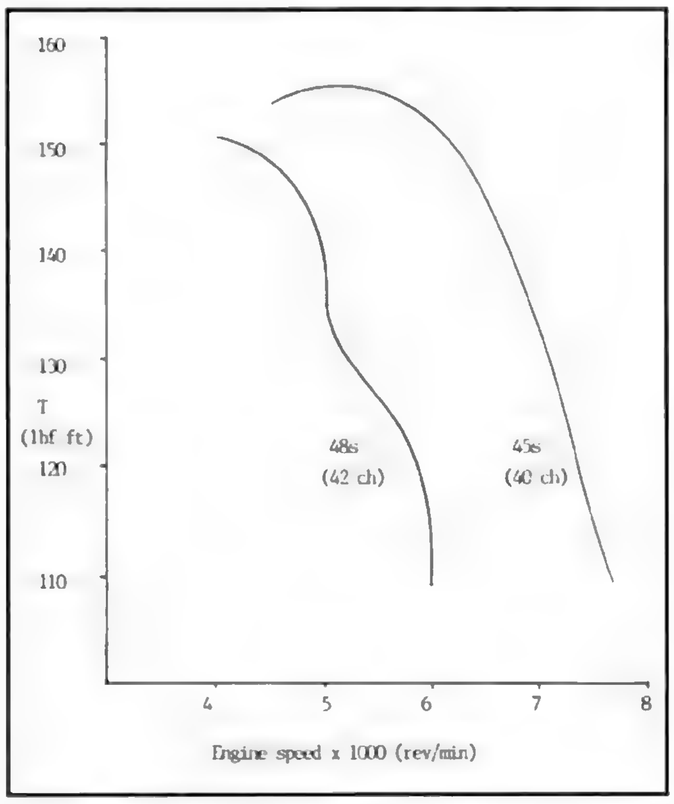
Full-load testing

At light throttle/low rpm the engine runs on the idle jets and it is not necessary to 'load' the engine to establish the correct jetting. Idle jets should simply be selected and the idle mixture screws adjusted to give optimum rpm. This is done at the lowest rpm at which the engine will idle with a virtually closed throttle (usually 750–1000rpm).

From approximately 3000rpm to maximum allowable rpm, the engine must be run against a load, either from a bench dyno/rolling-road, or on a test track/road. The purpose of this is to establish the correct jetting for the engine with the *throttle fully open*. Clearly, the load against which the engine can pull will vary either side of maximum torque, but the intention of testing with full throttle is crucially important since at full throttle (at any given engine rpm) the airflow through the carbs is at a maximum, and it is for this condition only that the jets must be selected. Apart from the idle condition, do not attempt to 'jet-up' the carbs at part-throttle.

Principal choke/jet functions

Idle jet – mixture at light throttle (generally 750 up to about 3000rpm).
Progression holes – mixture between idle jets and mains as throttle opens. The progression circuit can only be changed by blocking off holes (to lean) or extra drillings (to enrich), but the choice of idle jet emulsion tube helps determine the mixture quality during this phase.
Main jet – mixture at partial throttle/low rpm (approx 2000–3000rpm) and full-throttle mixture from 3000rpm upwards.
Air corrector jet – main power band mixture from approx 5000-max rpm and particularly 'top-end' mixture – (full-throttle) top ⅛ of rev-band.
Emulsion tube – blends fuel from main jet with air from corrector jet to produce emulsified mixture to feed secondary choke.
Primary choke – causes air velocity increase and pressure drop to allow fuel to bleed from jets into carburettor barrel. The major influence on the amount of bhp and torque.
Secondary choke – high airspeed in this choke draws fuel from main jet. Not normally necessary to change this from standard fitment.
Pump jet – its purpose is two-fold: primarily to pump neat fuel into manifold when throttle is opened suddenly; secondary function is to act as a high-speed enrichment jet to supplement work of the main jet at high rpm.
Float needle valve – controls the feed of fuel into the float chamber. The fuel demand of the main jet must be satisfied by the needle valve and fuel delivery from the pump. However, whilst additional fuel can be supplied by raising the fuel pressure from the pump, this may cause flooding at low rpm when the needle is lifted off its seat; from this point of view it is better to increase the flow of fuel by fitting a large-diameter valve.



10/32: Carbs too big? Forget actual torque figures as such, this is a back-to-back test on the same engine, conducted on the same manually-controlled dyno on the same day, and its importance is comparison between them. Engine comprised 10.2:1 CR, GC 3D cams. 42.5/37 race valves and was first run on 48 DCOE carbs, producing alarmingly low torque output shown. After much checking (compressions, cam timing) it was assumed that low air velocity in big carbs was responsible. Carbs were swapped for 45s and result speak for themselves. (Absolute torque figues were misleadingly high, but comparative result is valid enough.)

Carburettor balancing and initial setting-up

Some sidedraught variants have adjustment screws fitted, which allow the individual barrels to be balanced against each other. This feature is also common to IDF, DRLA, DCNF and some DCOE variants. These balance screws allow air to bypass the throttle plate, therefore their setting will affect the idle mixture. **(10/33)**

Referring to the diagram on the next page, proceed as follows (the procedure is similar for DCOE, IDF, DCNF, DRLA):

Disconnect throttle linkage from main lever (1).

Withdraw idle speed screws (2) from contact with main lever quadrant (3).

Unscrew the balance screw (4) from the balance quadrant (5) and with light pressure on levers (1 and 5) ensure both throttle plates are shut.

Hold the throttle plate shut and screw in the balance screw (4) until it just contacts the main lever, but does not cause the rear throttle plates to open. Then screw in the idle speed screw about 1½ turns to open both throttle plates.

Unscrew the idle mixture screws (6) three turns out and check that, where fitted, the idle air bypass screws (7) are screwed fully closed. (If a four-column mercury manometer is being used, connect to vacuum holes at (8) by means of adaptors.)

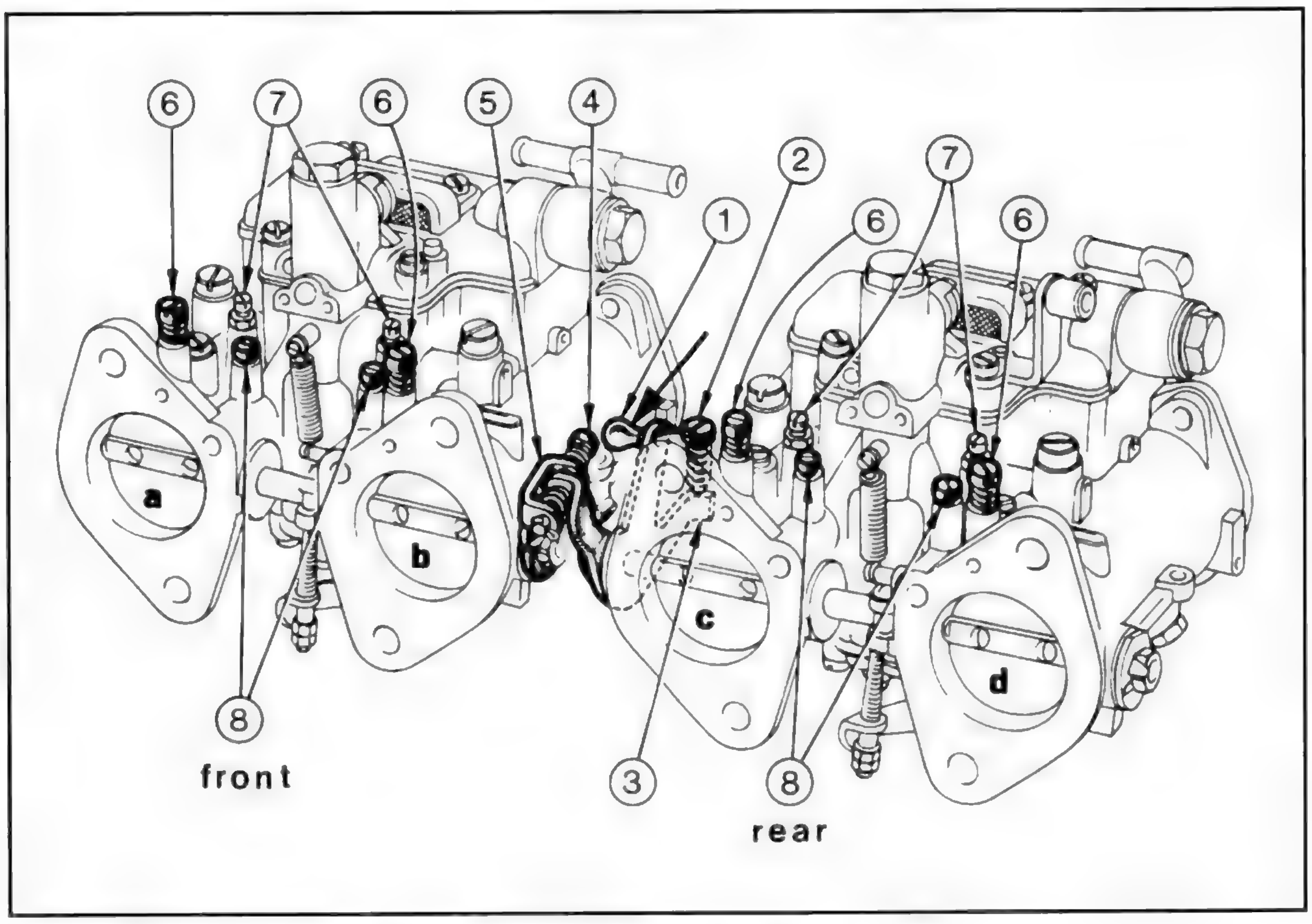
Note: The idle air bypass screws are

usually quite accurately set at the factory – do not adjust them until the balance has been checked with the manometer or balance meter!

Make sure there are no air-locks in the manometers (if used).

Reconnect the throttle linkage (ensure it has at least 2mm free play, *ie* the throttle cable is not holding it open). Pump the throttle three or four times to prime the engine.

Start the engine and run it at around 2000rpm until at correct operating temperature (it is a good idea to close the bonnet to assist warm-up). If the engine 'spits' then one or more of the idle mixture screws is too lean (assuming there are no bent valves!) – causing combustion to be so slow that it allows the flame to burn in the inlet tract when the inlet valve opens. Remember that optimum idle speed (around 850rpm) corresponds with correct idle mixture. The use of a tachometer to determine this is useful. Gunsons Colourtune is a handy little



10/33: Key components for synchronization and adjustment of a pair of Dellorto DHLA F40 carbs.

GCT TABLE OF SUGGESTED CARB JETS/CHOKES ETC

Use this data as a starting point – always confirm by road or dyno testing (Std – refers to jets fitted on new carb – refer to dealer if in doubt.) (See also *Case Histories*.)

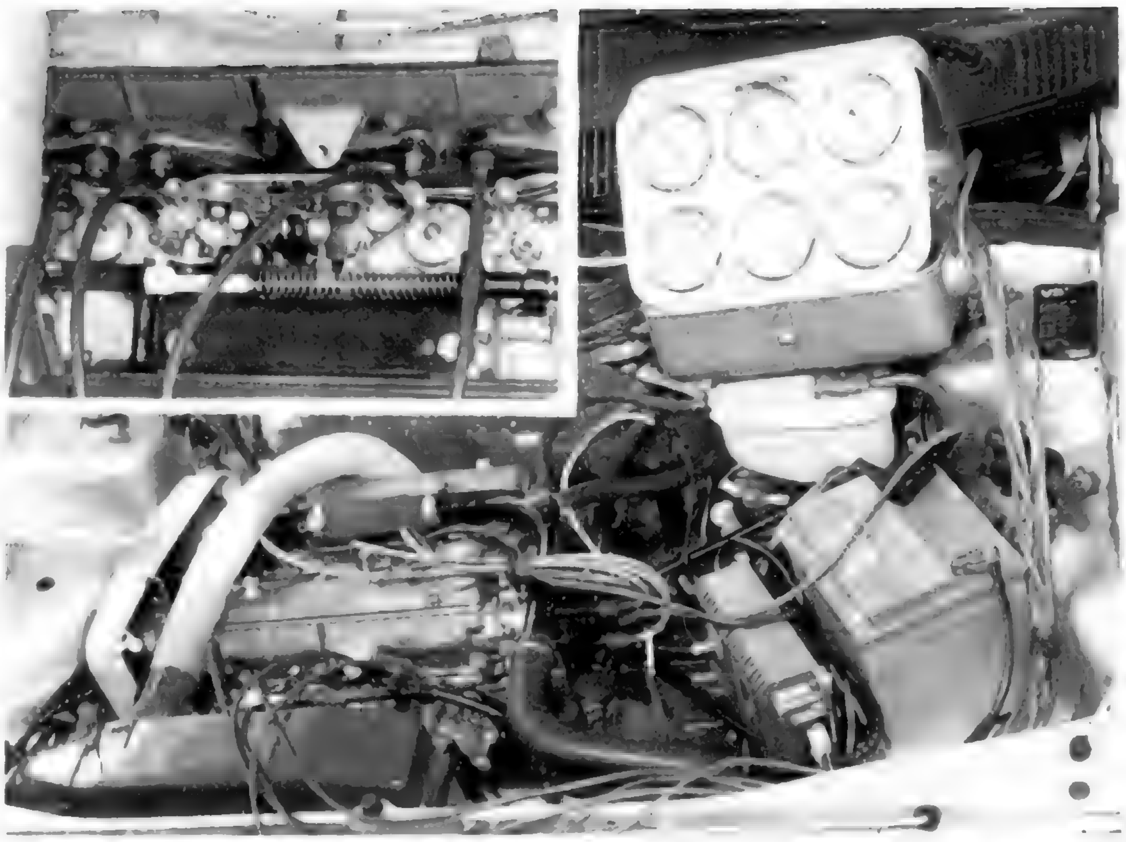
Engine	Carb	Choke	Main Jet	Emulsion Tube	Air Corrector	Idle Jet/ Holder	Pump Jet	Needle Valve
A. 131 2*l* 'Fast Road' (Std cams) (Approx 140bhp)	2x40 DCOE	30 32 34	120 130 140	F11 (DHLA No 5) F16 F16 (DHLA No 6)	180 175 170	45F9 45F9 50F9	Std Std Std	Std Std Std
B. 131 2*l* 'Fast Road' (Std cams) (Approx 155bhp)	2x45 DCOE	36	145	F16 (DHLA No 6)	170	55F8 DHLA 60.1	45 (Std)	Std
C. 131 1600 'Fast Road' (Approx 120bhp)	2x40 DCOE	30 32	115 125	F11 (DHLA No 5)	200 190	45F9 50F9	Std Std	Std Std
D. 131 1600 St II (GC 3A cams) (145bhp)	2x40 DCOE	34	125	F11 (DHLA No 5)	180	50F9	Std	Std
E. Standard 131 2*l* (126bhp)	2x45 DHLA	36	142	No 5 (F11 DCOE)	180	55.9	40	170
F Standard 131 2*l* (115bhp)	2x40 DHLA	32	145	No 5	180	55.9	40	170
G. St II 2*l* 131 GC 3A cams (178bhp)	2x45 DHLA	38 40	145 150	No 6 No 6	185 170	50.9 55.9	45 50	170 170
H. St III 1800 132 (44/38 valves) 320° cams (172bhp)	2x48 DHLA 2x45 DCOE	40 36	155 145	No 6 F16	170 170	55.9 55F8	50 45	170 170
J. St II Lancia Beta 2*l* (165bhp)	2x40 DCNF	32	165	F24	190	55	Std	Std
K. St III Beta 2*l* (198bhp)	2x48 DCOE	42	170	F16	180	55F8	50	170

Note: For more guidance of the selection/alteration procedure for jets/chokes and carb maintenance, the author recommends the excellent Haynes *Weber Carburettors Owners Workshop Manual* (ISBN: 1 85010 061 6).

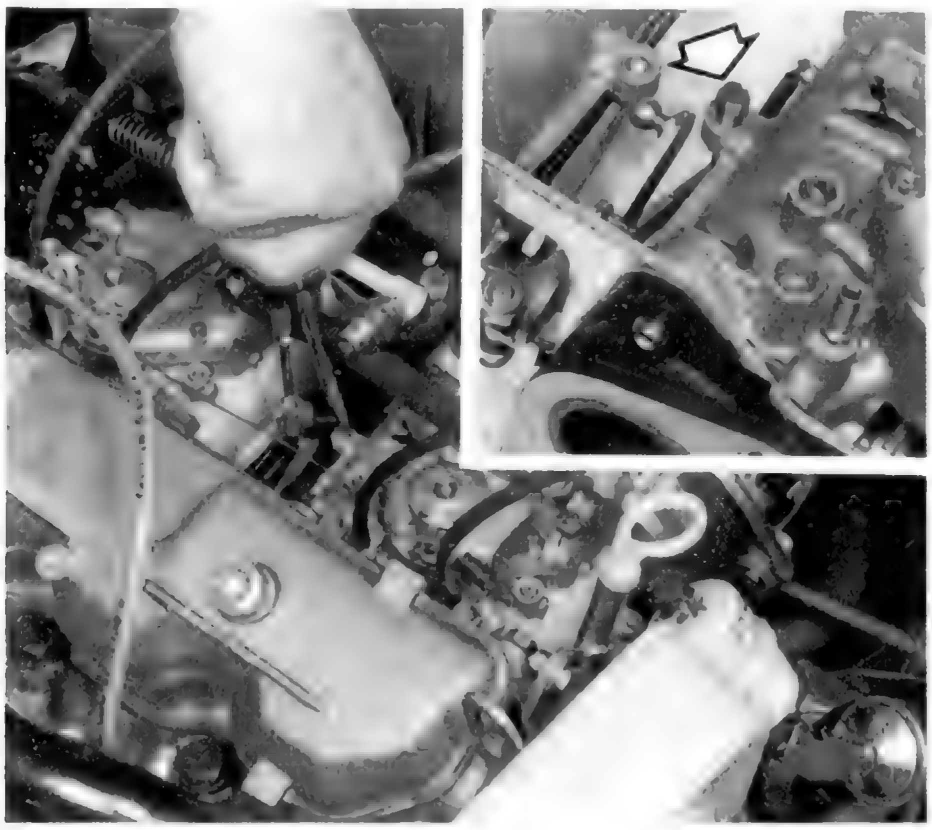
device which substitutes for a spark plug and allows the combustion flame to be examined – it should be 'royal blue' with an orange tint around the edge of the Colourtone window. (If there is any doubt about the ignition timing, check it with a strobe at this stage.)

Using either the manometer or a balance meter, measure the airflow in each choke. A low reading indicates either that an idle balance screw is too far closed, the idle mixture is too lean, or there is an air leak in the inlet tract (spray it with aerosol oil to seal the leak and the speed will pick up). Adjust the idle bleed screws (7) until good balance is achieved. If one carburettor is drawing consistently more than the other, adjust the balance between the pair by means of the balance screw (4).

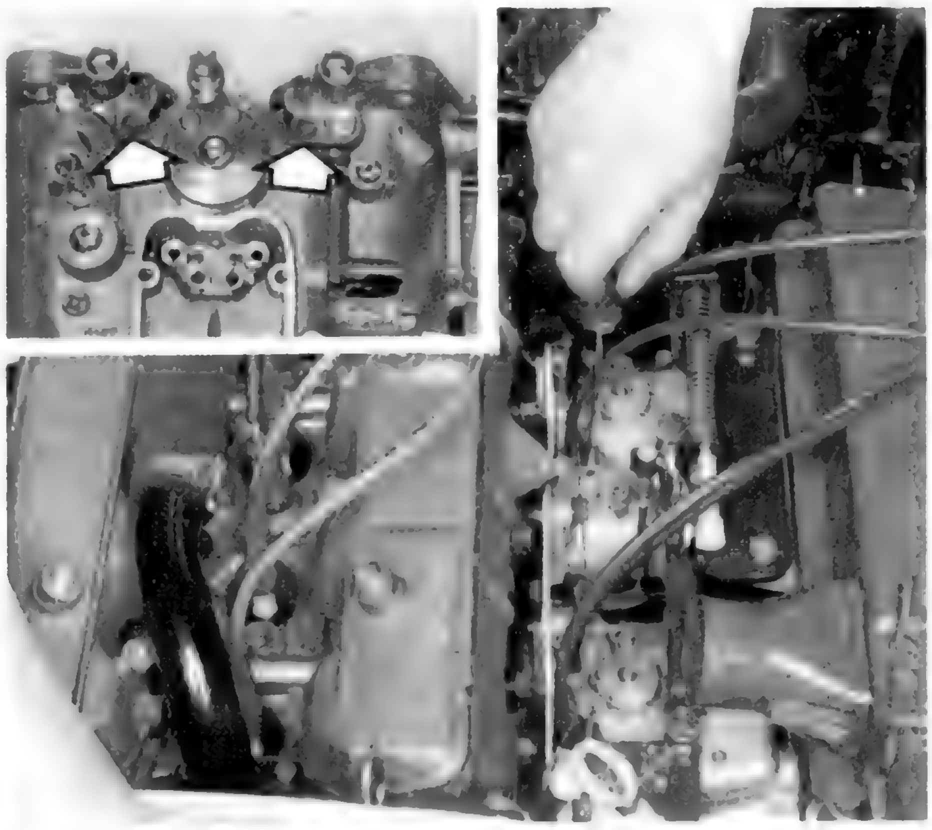
Finally, confirm that the idle mixtures are correct: screwing them in leans the mixture, out causes enrichment.



10/34: Idle adjustment on vehicle with engine warm, air filter fitted and choke completely in. Assuming accurate rev-counter is installed, disconnect four oil vapour recovery pipes from pickups and connect vacuum meter tubes… (engine is 130 TC)



10/35: …Using butterfly opening adjustment screw bring engine to speed of approx 850rpm. Check that vacuum gauge needles are all in same position…



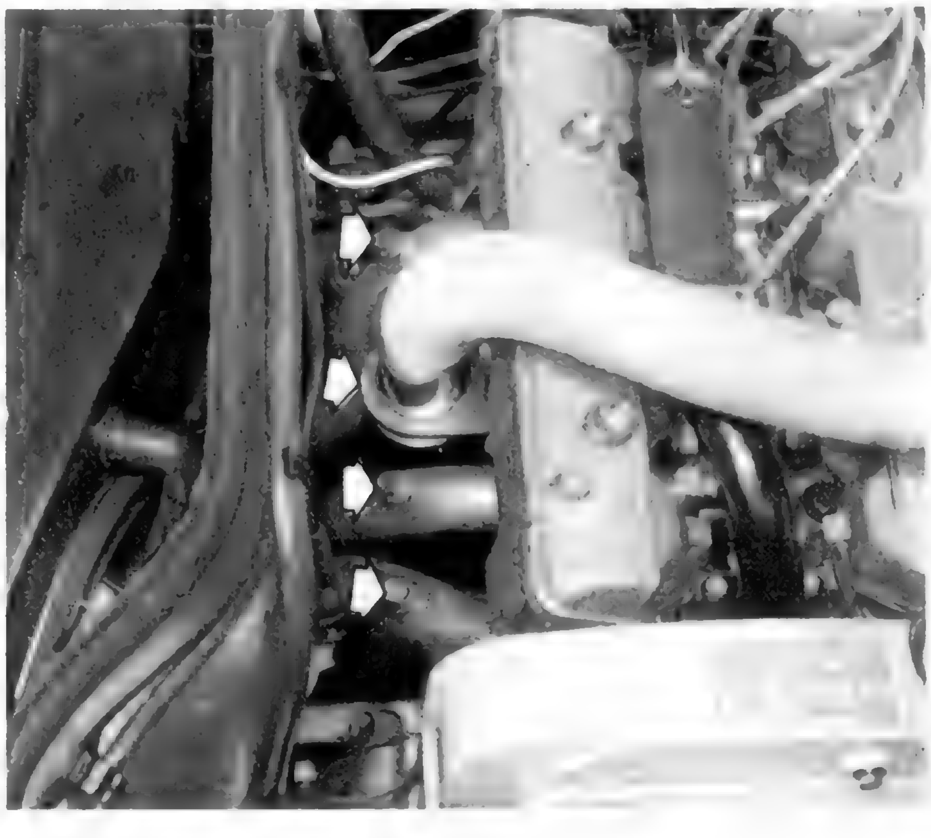
10/36: …If values for ducts in same carburettor are different, adjust air bypass screws (arrowed) until values correspond, repeating operation for other carb if necessary…

Other techniques

To set the idle mixtures most accurately requires unions to be fitted into the primary pipes and the CO level measured with an infra-red analyzer. (10/37)

Some engine analysis equipment is fitted with a power-balance facility. This allows the balance between cylinders to be checked by cutting out each cylinder for a predetermined time and counting the drop in engine revolutions. (GCT's quick trackside method is to pull off the plug leads one at a time and listen to the rev drop – at some personal risk, it must be said!)

CAUTION: Never look directly into the carburettor chokes when the engine is running – a misfire can cause flame to expand out of the inlet tract. If there is concern that a carb is flooding, inspect the choke barrels with a small mirror.



10/37: …Introduce exhaust gas analyzer sensor into appropriate section of exhaust manifold. (Illustrations 10/34–10/37 Fiat Auto SpA – copyright reserved)

10/38: Richard Ellis owns this Class 6 Autograss Lancia 1800/Fiesta. Autograss cars this well prepared are few and far between. Use of Facet Red Top pump to feed 45 DHLA carbs definitely requires use of pressure regulator (visible above carbs). Machining work and engine preparation by Mark Maynard of J & M Engineering led to output (with standard cams) of 135bhp @ 6000rpm, 125lbf ft torque at 4700rpm (9.8:1 CR, ported, 42/36 valves) – nice result on a standard-cam 1800.

Full-load checks
Main jets/chokes/emulsion tubes/pump jets/needles

In order to carry out this phase, a dynomometer or test track/road is required. With the engine warmed-up, load it up at full throttle at various speeds; bearing in mind that the torque (and hence mixture requirement) varies.

If jetting is established by road-testing, the engine should be run at full throttle at various speeds (*eg* steep hill, third or fourth gear, 3500–5000rpm and fourth gear, 6500–8000rpm on a flat road) for two minutes, then the clutch engaged, the engine cut immediately and the car coasted into a suitable parking area. A plug test can then carried out (carry swap jets with you!).

Too rich – plugs are coated and black (5½%–plus CO).

Correct – outer ring of plug thread is between pale grey and sandy brown and the insulator nose is a pale yellow/sandy brown colour (4½–5½% CO).

Too lean – outer ring is clean or very pale grey/white. Insulator is white (less than 4½% CO). Possible blistering.

If low-speed (3500–5000rpm) mixture is wrong, change main jets accordingly.

If top-end (6500–8000rpm) is wrong, change air correctors. Remember that this jet bleeds air into the emulsion tube so a bigger jet gives a leaner mixture.

If engine power does not feel adequate, or a correct mixture cannot be achieved, the chokes may be wrong. A large choke tends to give good top-end power, a smaller choke may give a better bottom-end torque. If the mixture runs consistently over-rich, whatever jets are used, the choke size is probably too small (for example a fast-road 2*l* – see above – with a CR of 10:1 would produce optimum power and torque with a 36mm choke, but when fitted with St II or St III cams, it would require choke diameters of 38–40mm).

Dyno or rolling-road testing will usually give similar results, with the added advantage that mixture can be optimized *versus* power output at various full-throttle engine speeds. It is crucial during this type of testing that the engine is not overheated at any time.

With large main jets, small air correctors and a compatible choke size, if the mixture cannot be optimized within the 3500-max rpm power band, larger needle valves must be used, or a higher capacity fuel pump. (Make sure the fuel delivery pipe is at least $\frac{5}{16}$" bore.)

If the engine suffers a serious flat-spot in the changeover rpm between idle and main jets (around 2000rpm), larger pump jets are needed, though this condition can be created by using idle jets which are fundamentally too small and the idle mixture screws are on the limit of enrichment.

Remember: Incorrect mixture damages your engine – perhaps terminally! Bore washing can damage rings/bores and contaminate oil, thus degrading oil performance and leading to damaged bearings, guides and oil pump within a matter of a few hours' use!

If you overheat the engine, the head gasket will blow.

Tests prior to running-in and after are *both* required.

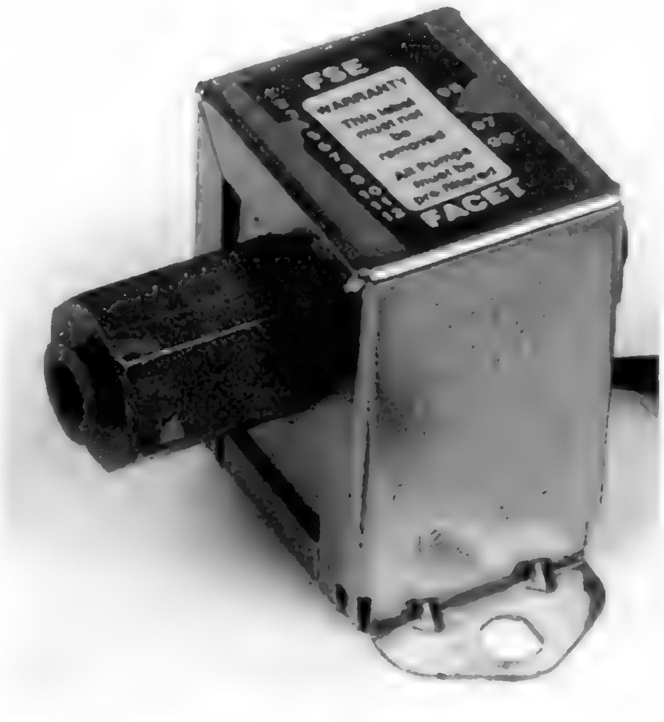
Fuel pumps

A twin-carb installation (on all the aforementioned carbs) can cope quite adequately with about 6lbf/in^2 fuel pressure, provided the float needles are in good condition. The line pressure will vary from maximum pump output (engine switched off – fuel pump running) to 'running pressure', which will depend on the fuel demand from the engine. Unfortunately, a high flowrate tends to mean high pressure where fuel pumps are concerned, and a full-race normally-aspirated TC may demand as much as 5lbf/in^2 line pressure to supply the requirement of 2x48 DCOEs on full power (around 8 race miles per gallon). In this respect, a high-output (flowrate) pump may need a regulator to ensure that at low rpm the float needles are not lifted off their seats. **(10/38)**

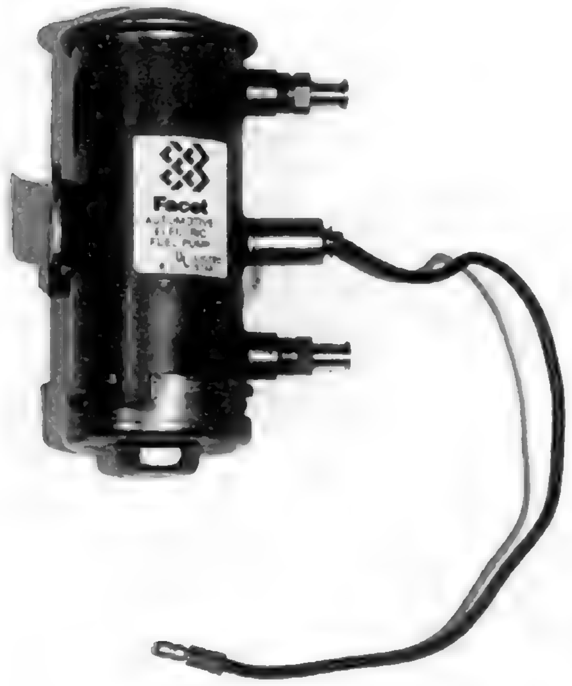
An alternative for DHLAs is 'turbo' needle valves, which withstand higher fuel pressure because of the pressurized float chamber on 'blow-through' turbo/DHLA set-ups.

An electric pump is preferable to a mechanical one since it can be activated to prime the carbs prior to cranking, thus reducing the load on the battery, which is especially important on cars not fitted with an alternator. Additionally, there is a risk with the standard fuel pump, if the diaphragm becomes punctured, of fuel leaking into the sump, causing chronic oil contamination.

The most popular electric pumps (which have always been used and recommended by GCT) are Purolator (formerly Facet). **(10/39, 10/40)**

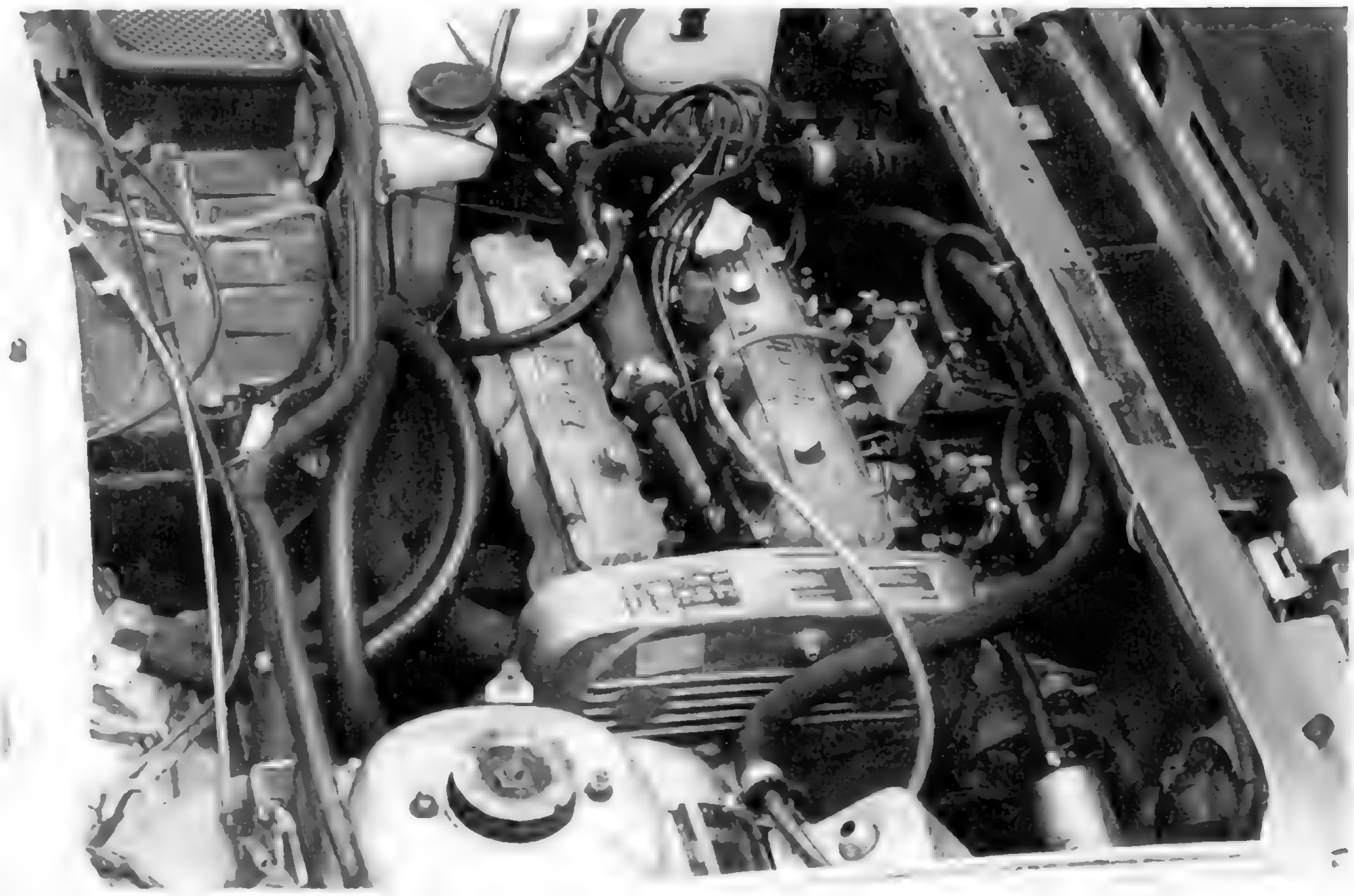


10/39: Facet solid state pump.



10/40: Facet interrupter pump.

10/41: Gordon Ingle and car!



10/42: Gordon Ingle, a principal scientific officer at the Defence Research Agency in Bedford, owner of a Lancia Delta equipped with a GCT 1600 motor – 9.6:1 CR, ported, 43½/36 valves, standard cams – writes:

"Dear Guy. Here is some information about my car. I won't list the spec, since you will know it as well as I do, except to confirm it has got sidedraught Dellorto DHLA 40s. It was something of a struggle to fit these carbs."

[Author's note: The GC straight-shot inlet manifold would have helped here – being nearly 1" shorter. Unfortunately, at the time, we didn't realize it could be machined from the Monte Carlo casting!]

"Eventually, I had a non-standard radiator made, to my spec, which is less deep, but wider than the standard one. It is fitted lower down in the grille aperture and allows room for the carb airbox (my own design) to extend forward above the radiator and draw in cold air through the grille.

The oil cooler (13-row) is fitted behind and slightly below the front crossmember, making it slightly vulnerable I suppose, though I've had no problems.

I'm very pleased with the engine, it has a very wide useful rev-range – you can accelerate away smoothly in 4th from about 1,200rpm, for example, making it very flexible. I travel about 36 miles a day, to and from work, and I get nearly 30mpg, which I think is pretty reasonable. It hardly uses any oil and has proved utterly reliable – and it makes a lovely noise! The latter is helped a little by the exhaust system I got from Italy.

I would certainly endorse your advice to keep under-bonnet temperatures down; after a long run temperatures could get too high and it was quite noticeable how the power suffered. [Author's note: see also Chapter 13.] I used the louvres in the rear of the bonnet to exit the engine bay hot air. This necessitated removing the blanking plate [these louvres are cosmetic on the 1600 GT], then fitting an air scoop on the underside of the body, since the louvres are the wrong side of the bulkhead, if you see what I mean. This requires shaping the bulkhead to the contour of the scoop, as well as moving the header tank over to the other side of the bay.

This mod has proved pretty successful. Current work is to fill in the gaps between the radiator and the bodywork to force all the air through the cores – during the recent hot weather, cooling temperatures still got a little high when the car was operating in town traffic conditions.

Jonathan Douglas of ITG has been very helpful – his latest assistance has been to donate some foam filter material, which I inserted between the grille and the front of the carb airbox, really just to intercept the larger debris (flies etc) from entering the main filter, which is mounted in the mouth of the airbox. A useful side effect of this extra filter seems to be to remove all carb icing problems. Previously, at just above freezing conditions, the engine could run rough and cut out at idle. I hope this is helpful."

[Author's note: 3A profile will take this engine to 145bhp.]

Fuel return lines

A fuel return line is usually incorporated into production models where a mechanical pump is fitted, to keep the fuel cool by circulating it back to the tank, thus preventing any possible vapour build-up in the feed line caused by a combination of low (suction) pressure and high atmospheric temperature acting on the pipe. With fuel pumps located at the tank, this line can be blanked-off. If the return line is retained it is vital that the delivery to the carbs is not compromised (on full power) as a result.

Monitoring fuel pressure

Because chronic detonation damage can occur almost instantaneously if the fuel delivery is disrupted at full-throttle operation, it is well worth installing a fail-safe fuel pressure monitoring system to protect the engine. This could comprise a low-pressure warning switch in the fuel line (between pump and carbs), which would automatically isolate the engine ignition if the line pressure (running) dropped below a preset level *(see Pump data – running pressures)* – a 0–10lbf/in² fuel pressure gauge is available from Raceparts. Alternatively, a dashboard-mounted fuel pressure gauge or fuel pressure warning light is highly desirable.

Another method of ensuring a reliable fuel supply is to fit twin fuel pumps. This increases the flow; to set the required pressure a regulator will be needed. A fuel pressure gauge will be needed to calibrate the regulator in order to achieve the required 6lbf/in² maximum static pressure (to prevent flooding).

PUMP OUTPUT DATA (all electric pumps are 12v neg earth)				
TYPE	FLOWRATE (gall/min)	DRY PRIME (in)	STATIC PRESSURE (lbf/in²)	RUNNING PRESSURE (lbf/in²)
131/132 2*l* (mech)	0.3 Use up to low St II only			2.8–4.3
130TC (elec)	0.5 Can be used up to n/a full-race (regulator fitted as std)			2.2–5.9
FACET PUMPS (elec) Solid state standard (Pt No 40105)	0.3 Use up to low St II only – mount near tank (no regulator required)	12	2.5–4	2.5
SOLID STATE 'Fast-road' (Pt No 40106)	0.42 Use up to high St II – mount near tank (no regulator required)	18	4.5–6	4–4.5
SOLID STATE 'Competition' (Pt No 40147/40077)	0.53 Use up to St III – mount F or R (regulator req)	48	7–8	5–6
FACET 'INTERRUPTER PUMPS' 'Silver Top' (Pt No 476087)	0.41 Use as 40106 – mount F or R	36	4–4.5	2–5
'Silver Top Comp' (Pt No 480530)	0.53 Use as 40147 – mount F or R (no regulator req)	48	5–6	3.5–4
'Red Top Works Spec' (Pt No 480532)	0.75 Use up to n/a full race – mount F or R (regulator req)	48	6.5–7	5

Note: The Facet range of pumps are marketed in the UK by Peter Huxley, of Fuel System Enterprises (brother of Terry Huxley of Datum – carburettor specialists, which is very convenient) in kits comprising pump, mountings and unions. FSE also supply an excellent range of fuel filters and regulators.

CARBURETTOR PROBLEMS	
SYMPTOM	POSSIBLE CAUSE/CHECK
Engine will not start	– No fuel in carb. Remove float chamber cover to check. Check fuel pump delivery. – Blocked idle jet or fuel gallery. – Engine flooded due to excessive pumping of throttle (pump jets pump fuel into manifold) or flooding due to incorrect float level, faulty needle valve or excessive fuel pressure/carb misalignment. If flooding is suspected, switch on fuel pump (do not crank engine due to fire hazard – use a mirror to inspect carb barrels for flooding if a mechanical pump is fitted) and see if fuel is dripping into chokes. Also inspect plugs to see if they are 'wetted-up' (problem may be a weak spark or poor compression).
Uneven idling	– Idle mixture too rich (plugs will be sooted-up) or too lean (spits back or 'pops' on overrun). Check idle jets/idle mixture screws. – Carbs out of balance. – Air leak on manifold or throttle spindle. – Choke device not disengaged. – Contaminated fuel (remove top cover to check fuel quality). – Idle jets loose, bypass screw loose. – Throttle plate sticking (fuel bleeding from progression holes).
Poor progression (idle OK) under low-throttle acceleration	– Idle emulsion tubes too rich/lean (a CO check will indicate which way to go!).
Poor performance, quarter-full throttle	– Blocked main jet/fuel gallery. – Wrong chokes – too small and an over-rich mixture may result, too large and the mixture may not enrich satisfactorily. (Also the air velocity may be too low to give good torque.) – Main jet too small (lean). – Main jet too large (rich). – Emulsion tube wrong (GCT have only ever used DCOE F11/16, or DHLA 5/6 – *see table*). – Secondary chokes installed back-to-front or not secure(!).
Flat-spot under acceleration	– Pump jets too small, or fault with pump assembly/blocked jet.
Poor performance top-fifth rpm	– Air correctors wrong – too large – weak, too small – rich. – Throttle not opening fully. – Insufficient fuel delivery. – Needle valves wrong. – Blocked air corrector/fuel gallery.

Note, of course, that the various problems of faulty needle valve, faulty fuel pump, etc, can inhibit the production of power throughout the whole range. Always tackle fault-finding one step at a time – never change two components at once.

PART TWO: FUEL INJECTION

Background

The use of fuel injection in automotive applications was pioneered by Daimler Benz in the 1930s for use on GP cars, and its use on petrol engines achieved prominence through the adoption by Messerschmitt of fuel-injected Daimler-Benz aero engines as the powerplant for the famous ME109 fighter plane during World War 2. It was found (apart from its obvious benefit over carburettors in aircraft high-'g' situations) that owing to the greatly improved fuel metering/atomization (and thereby better distribution of the correct octane of fuel to each cylinder) and the removal of a requirement for a choke (thus reducing the work lost in pumping the charge into the cylinder), that a power improvement of around 10% *per se* was possible. Subsequently, in the 1970s a number of car manufacturers experimented with fuel injection systems – Mercedes, who continued to work with what were effectively in-line diesel injection pumps modified for gasoline use; Triumph, who adopted the Lucas 'shuttle pump' for the 2.5 PI (this was actually a distribution unit, the pressure coming from a separate fuel pump – later, a derivative was used to great effect on the Cosworth BD series of engines); BMW, who chose a Kugelfischer mechanical pump for the 2002 ti; and indeed Fiat themselves, who used a similar set-up on the works 131 16v Abarth, and a Bosch mechanical pump on the Lancia 037.

By the late 1970s, the race was well and truly on to develop a mass-production injection system; in Europe the lead was taken by Bosch, of Germany, who developed the outstanding Bosch K Jetronic (mechanical) and L (electronic) systems which were fitted to millions of production vehicles. Fiat/Lancia adopted the L Jetronic system for their (US) 2/124 Spider, Argenta *ie*, Trevi/Beta *ie,* and Weber-Marelli developed a system for the later models.

Fuel injection is popular because the improved octane distribution allows manufacturers to use higher CRs than would be feasible (for a given fuel rating) with carbs, thus raising the thermal efficiency of the engines; and improved metering/atomization reduces emissions, raises mpg and increases the working life of the engine. Extra power (the Argenta *ie* develops 12% more power than the 132 2/ on the same basic engine) comes almost as a by-product!

Bosch L Jetronic system

This system is shown in outline in **10/43**. The mode of operation is basically quite simple.

The fuel is pumped *via* a filter (and usually a pressure accumulator to help with restarts and maintain line pressure) from an electrically driven 'roller cell' fuel pump to the main injection rail, where it feeds the four TC injectors. The fuel amount injected is controlled in two ways.

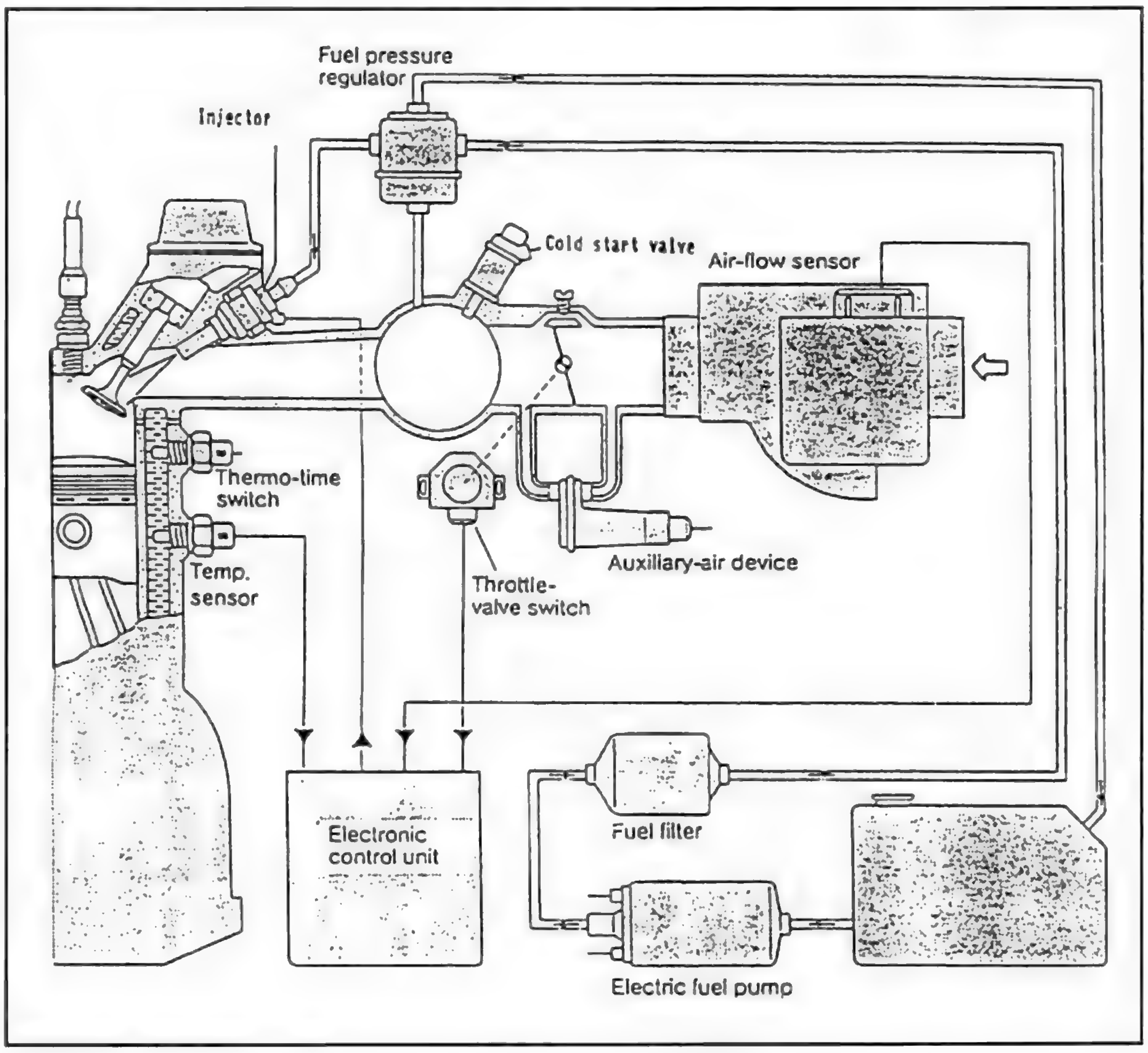
The injection pressure is controlled by the fuel pressure regulator, which monitors engine load *via* manifold pressure. At closed throttle, the manifold gives high vacuum (low-load condition) and the regulator returns a high volume of fuel to the tank; at open throttle, the vacuum condition is reduced and the fuel pressure is raised.

The duration of injection is controlled electronically by an electrical signal to the solenoid windings of the injector from the ECU (electronic control unit). Whereas with the early 131 Kugelfischer system, the only regulator device controlling the injection was a mechanical governor and a vacuum sensor, the injection control of the L Jetronic is supplemented by use of an airflow sensor. In this unit a spring-loaded flap valve opens as the airflow increases (when the throttle is opened). A potentiometer inside the unit transmits a signal to indicate the mass flow-rate to the ECU. Data from the engine coolant sensor adds to the picture.

On cold starts, a thermo-time switch can activate a cold start injector (though on some systems the injectors are held open longer to perform the same function) and throttle bypass valve (auxiliary-air device). The throttle switch supplies full-throttle enrichment under full-power operation. Injector timing is controlled by linking the distributor to the ECU – advance of ignition can be centrifugal (Beta *ie*), or fully mapped inside the ECU.

The 'special tuning' potential of this system is quite good, but because the inlet tracts are linked to a common throttle plate there is a loss of induction ram effect due to interference between cylinders compared with the one-choke-per-cylinder layout of a twin-carb system. The throttle response compared with race carburettors is poorer because of the lag between airflow sensor plate reaction and the increased injection requirement when the throttle is suddenly opened. In addition, the airflow sensor plate tends to impede the airflow.

The system can cope adequately with the simple tuning procedures of high CR,



10/43: Schematic of Bosch L Jetronic layout.

porting/blueprinting to the cylinder head, but a change of valve size or camshaft type requires the ECU to be remapped to increase the injector duration. Raising the fuel pressure by adjusting the fuel pressure regulator is not the complete answer since it may over-fuel during the acceleration (transient mode).

Companies involved in research on *other engines* utilizing this system have experimented extensively with larger throttle bodies, modified airflow sensors, etc, but for the TC, unless the race regulations specifically permit use of the system, and a sizeable budget is available for bench-dyno testing and mapping(!), this system is best only retained for fast-road use.

That said, a detailed analysis of the technical data associated with the L Jetronic is not strictly relevant to this particular book. However, the following fault-finding table may be useful:

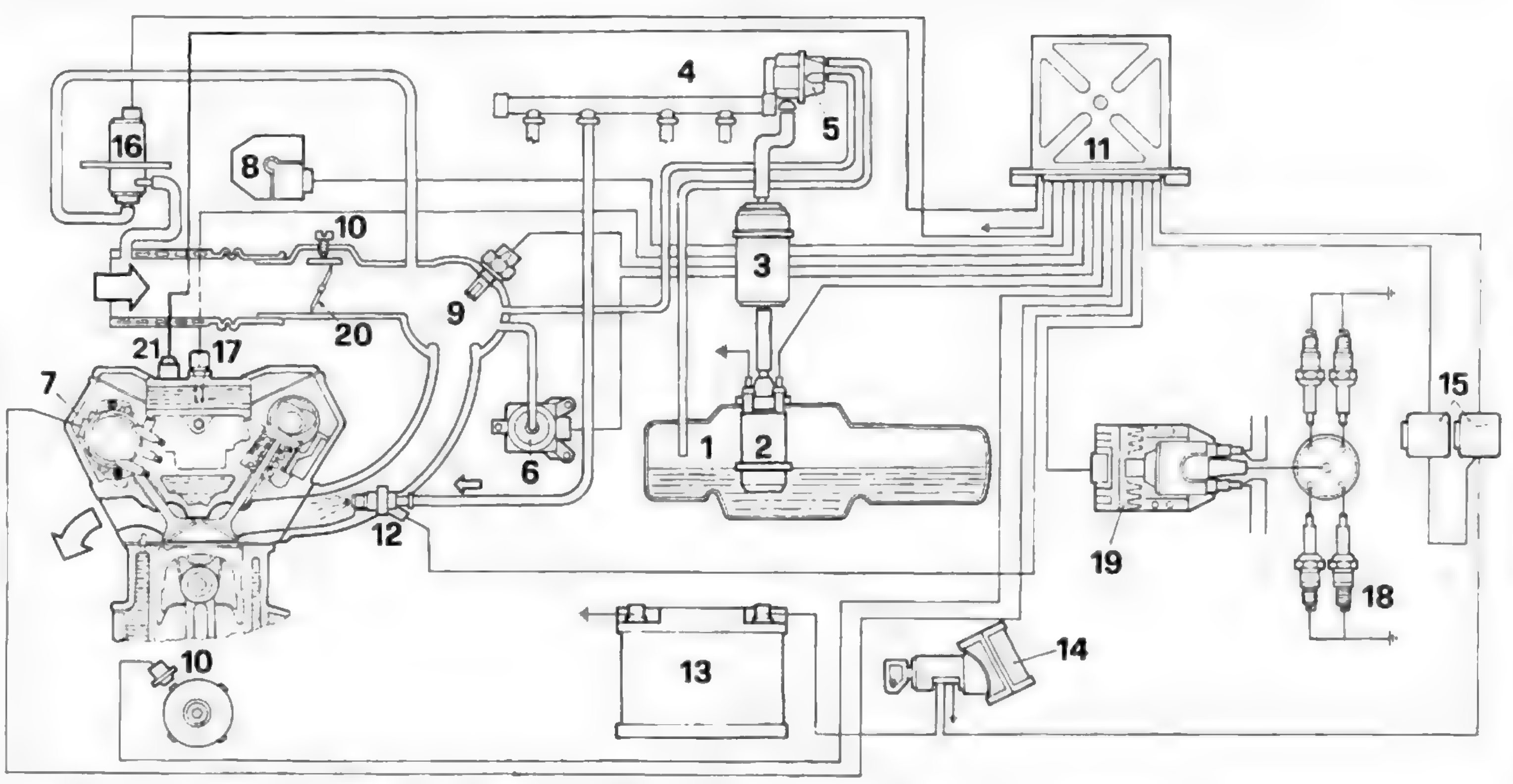
FAULT	POSSIBLE CAUSE/CHECK (in order of precedence)
Engine will not start (cold)	– Aux air device stuck shut – disconnect air hoses – engine may start. – No injection – remove injector and crank engine (hold injector in container to minimize fuel hazard). – Airflow sensor stuck – check free movement. – Faulty connection. – Faulty ECU or airflow sensor (replace as last resort).
Engine will not start (hot)	– Faulty aux air valve (stuck open). – Faulty fuel accumulator (low line pressure).
Erratic idle	– CO level wrong (ECU incorporates mixture screw). – Throttle plate incorrectly set. – Dirt/corrosion on airflow meter potentiometer (inspect via top cover). – Aux air device not closing – remove and inspect hot. – Leaking injector seal. – Faulty ECU.
Lack of power, excessive consumption	– Low fuel pressure (check feed pressure with fuel gauge). – Blocked fuel filter (approx 10,000-mile change interval). – Faulty or incorrectly set fuel pressure regulator. – Dirty injector (they must be cleaned ultrasonically). – Potentiometer problem as above. – Aux air device stuck open. – Defective fuel pump (or voltage feed too low). – Faulty connection. – Faulty ECU.

[*Author's note:* For an in-depth analysis of tuning the Jetronic, see the outstanding book *Bosch Fuel Injection & Engine Management* by Charles Probst – ISBN 0-83760-300-5 – available in the UK through Motor Racing Publications.]

Weber-Marelli system (OE vehicles)

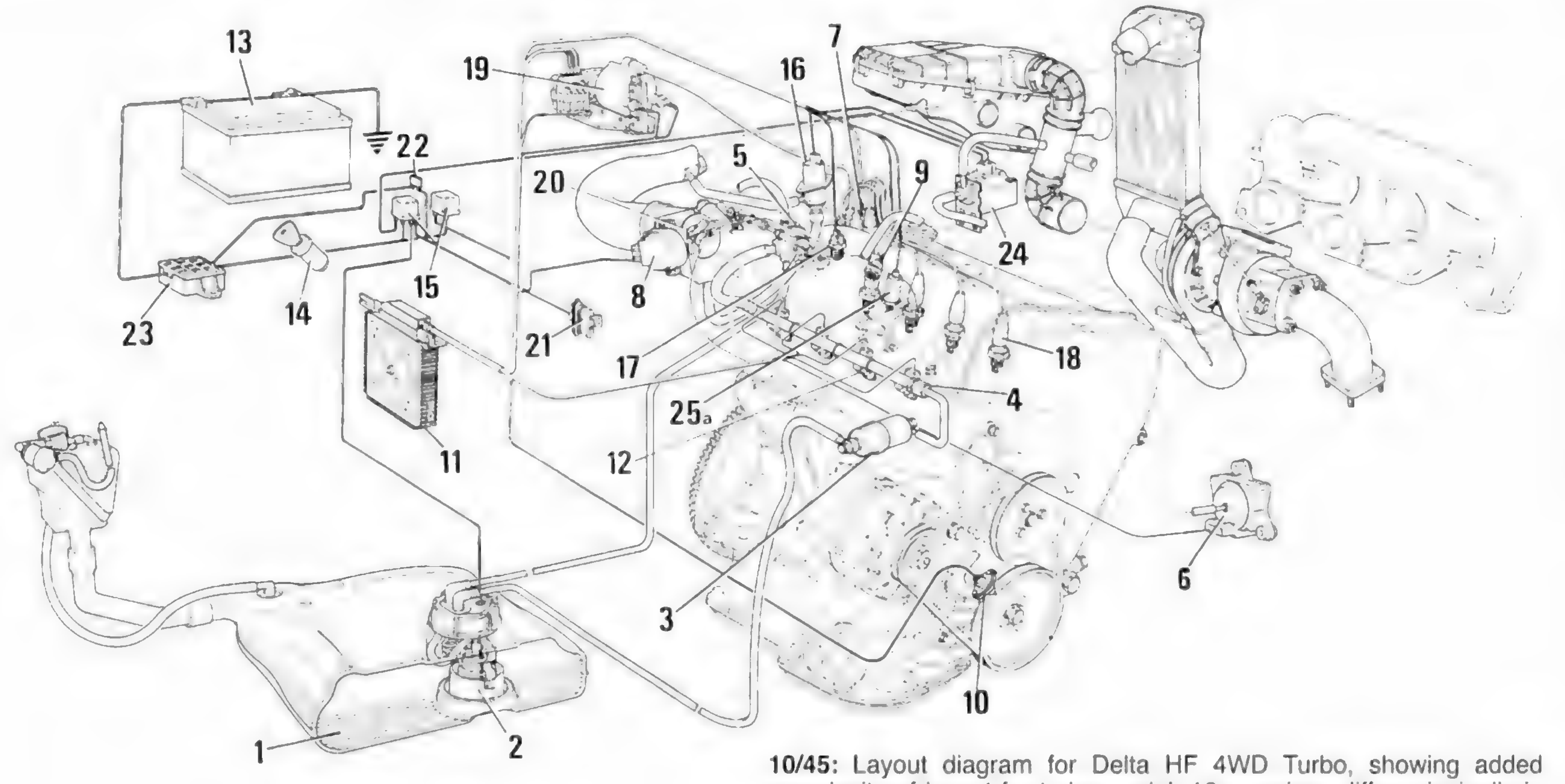
Derivations of this system are used on the Delta Turbo 1600 *ie*, Prisma/Delta HF 4WD, Thema *ie*/turbo and Integrale 16v. The injection differs principally from the L Jetronic in that it has no airflow sensor. It is fully mapped *(see later Weber programmable system) ie,* the fuel delivery is partly dependent upon the signals from the manifold pressure, rpm and throttle position (potentiometer) sensors to indicate to the ECU the load/speed condition under which the engine is operating. As with the L Jetronic system, injection rail pressure is controlled by the pressure regulator and injection timing/rate from the ECU. All systems have fully mapped ignition, controlled by the ECU for optimization of the power band. **(10/44, 10/45)**

The fully mapped systems are not overly airflow-sensitive, *ie,* they cannot differentiate between the standard airflow characteristic and the increased flow of a modified engine in a radical state of tune. It is, therefore, vital, with both normally aspirated and turbo versions, that if such tuning (big valves, cam swap, raised boost) is carried out, the settings of the ignition and full EPROMs (electronically programmable read-only memory) are checked on a rolling road or bench dyno. A modest increase in fuel pressure (the standard regulator can be adjusted – remove the blank plug) of around 2–3lbf/in^2 seems to work well with simple

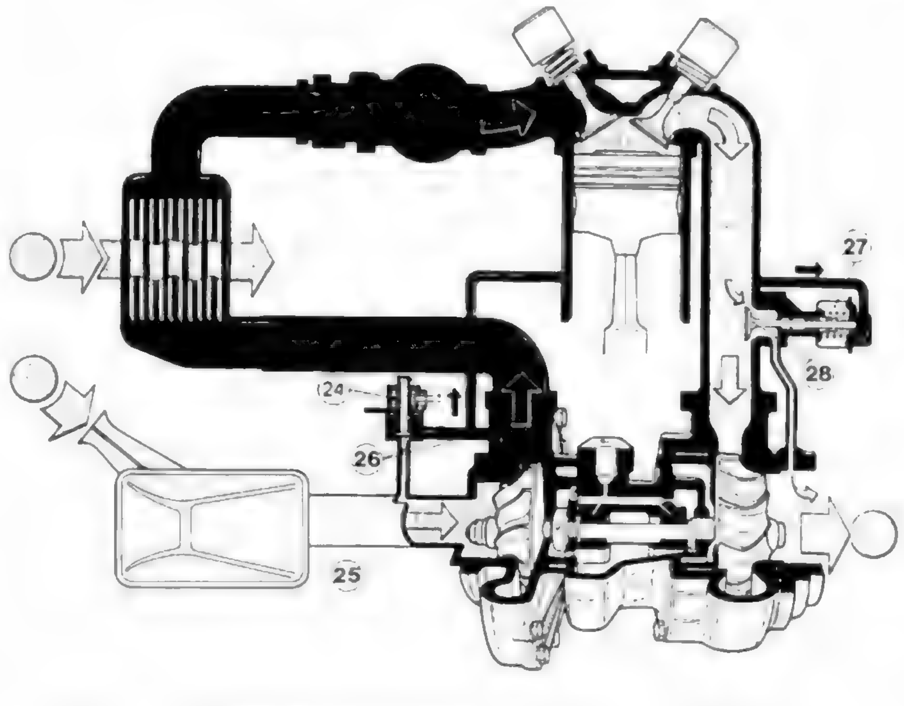


1 Fuel tank
2 Electric fuel pump
3 Fuel filter 4 Fuel manifold
5 Fuel pressure regulator
6 Intake air absolute pressure sensor
7 High tension distributor with injection timing sensor
8 Butterfly valve position sensor
9 Intake air temperature sensor
10 Rpm and TDC sensor
11 W.I.I. electronic control unit
12 Injector 13 Battery
14 Ignition switch
15 IAW injection/ignition system relays
16 Engine idle speed automatic adjustment supplemenary air solenoid valve
17 Coolant temperature sensor
18 Spark plug 19 Ignition unit
20 Butterfly valve
21 Anti-detonation sensor (for Delta HF 4WD turbo only).

10/44: Wiring layout diagram for Delta 4WD (2l). (Fiat Auto SpA – copyright reserved)



10/45: Layout diagram for Delta HF 4WD Turbo, showing added complexity of layout for turbo model; 16v versions differ principally in that there are individual coils for each spark plug. (Fiat Auto SpA – copyright reserved)

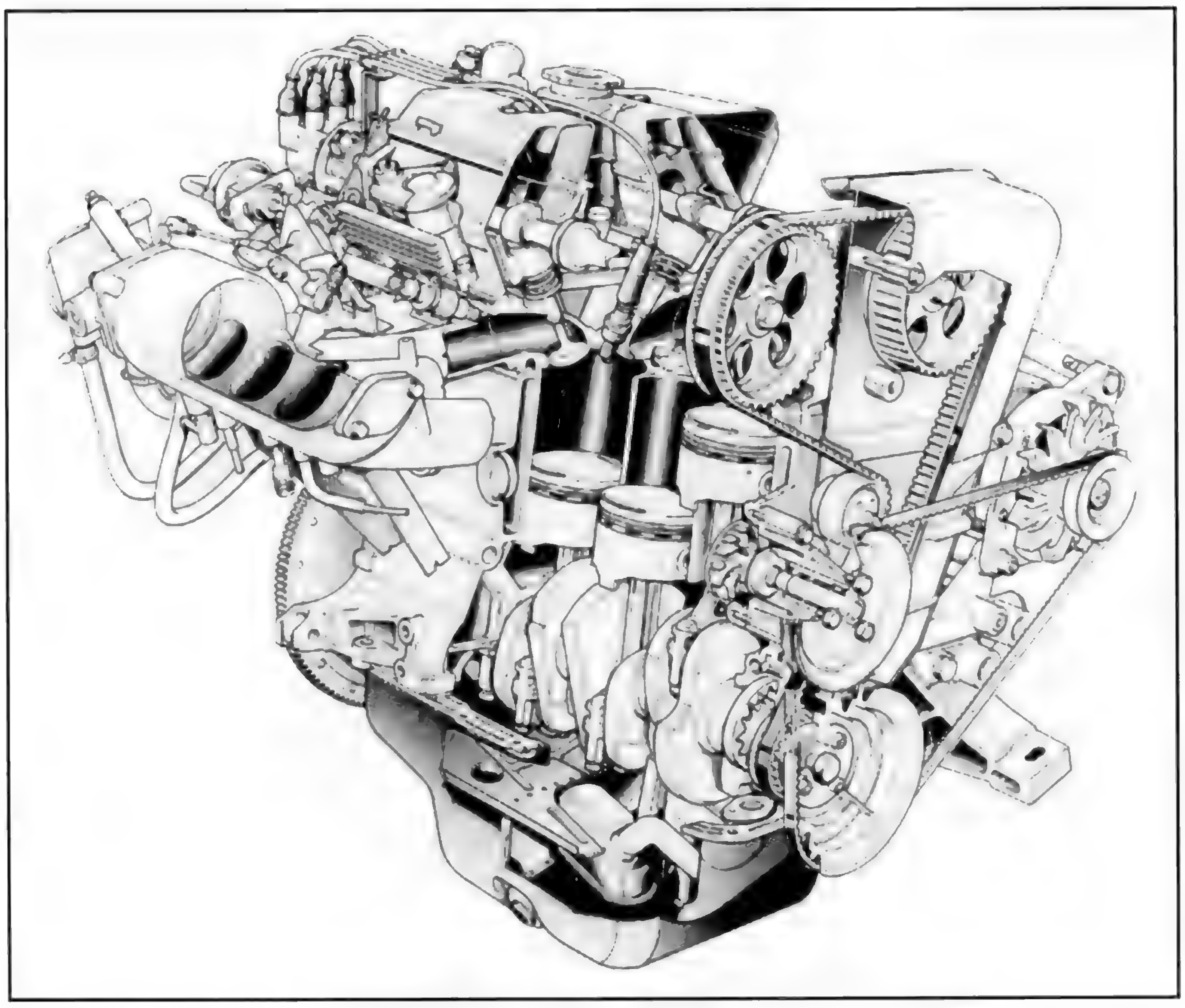


1 Fuel tank
2 Electric fuel pump
3 Fuel filter
4 Fuel manifold
5 Fuel pressure regulator
6 Intake air absolute pressure sensor
7 HT distributor with injection timing sensor
8 Butterfly valve position sensor
9 Intake air temperature sensor
10 Rpm and TDC sensor
11 Electronic control unit
12 Injector
13 Battery
14 Ignition switch
15 Injection/ignition relays
16 Additional air solenoid valve for automatic adjustment of the engine idle speed
17 Coolant temperature sensor
18 Spark plugs
19 Ignition unit
20 Butterfly valve
21 Diagnostic socket
22 Fuse
23 Connector
24 Over-boost solenoid valve
25 Air intake duct from the filter
25a Detonation sensor
26 Compressed air ducts from the turbocharger
27 Supercharging adjustment actuator
28 Wastegate valve

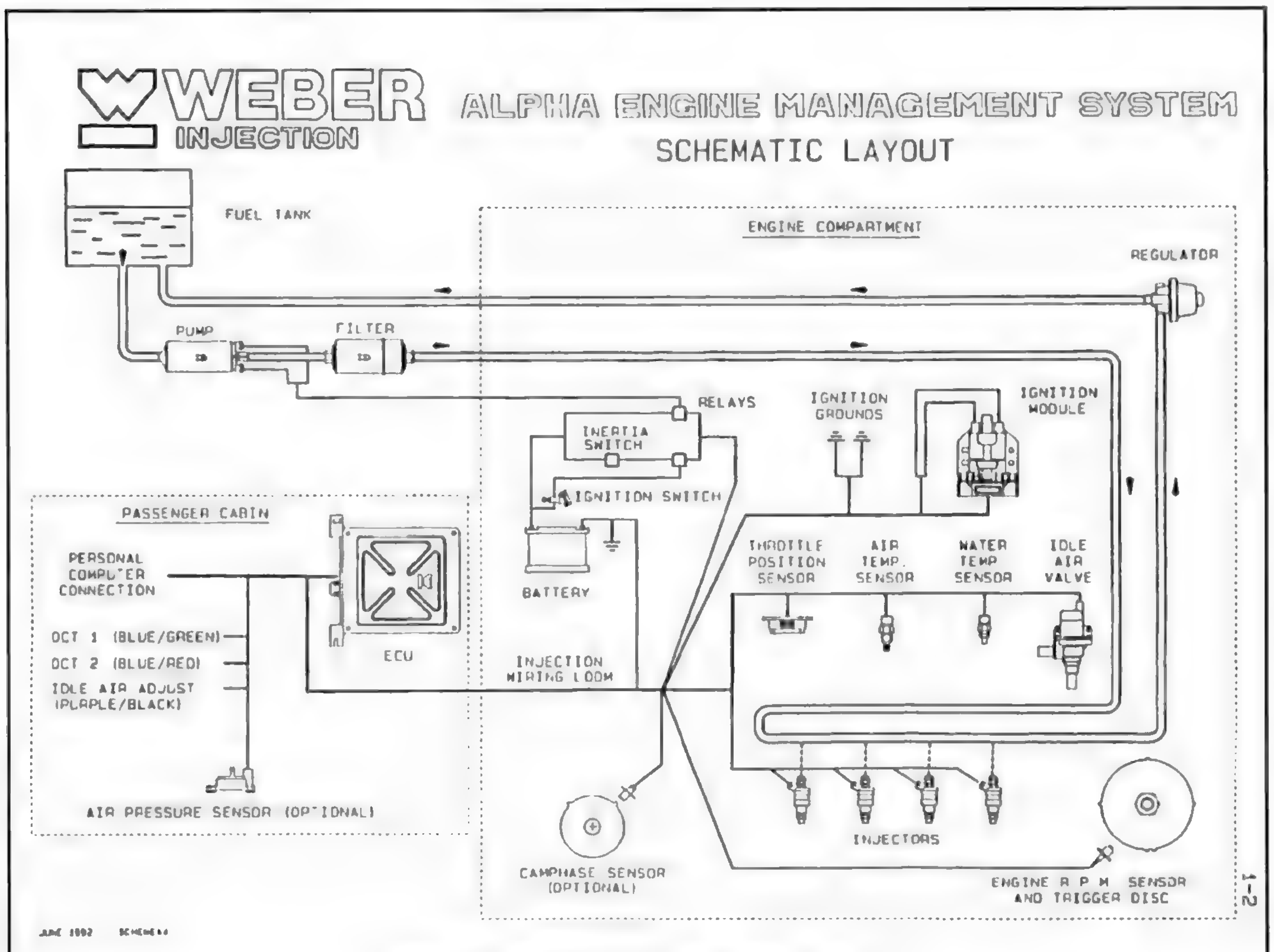
tuning, *eg* ported/blueprinted head.

It is worth noting that production models are 'chipped' for economy (and the possibility of low-grade fuel) and some benefit may be had by replacing the standard EPROMs with modified types; although GCT do not have any hard data to quantify this, power gains of around 5–10% would seem to be reasonable – if the mapping is carried out comprehensively.

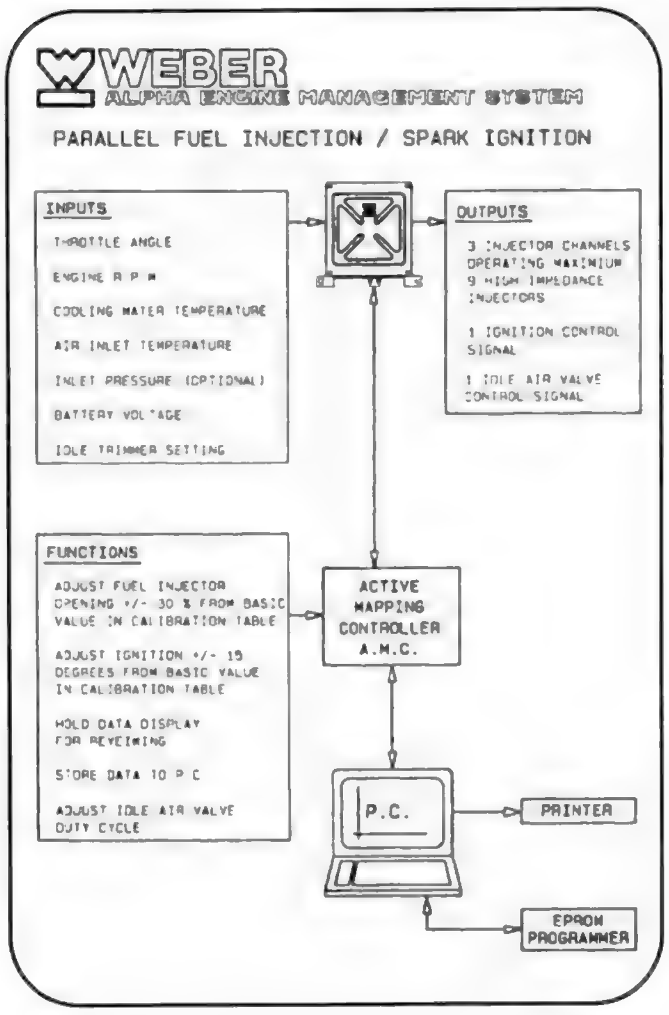
The diagnostic sequence for fully mapped engines is far too extensive to list here (and there are considerable variations between models – which are constantly changing, it must be said!), but essentially it starts with simple checks of connections and fuel pressures (as with L Jetronic), moving through component functions to finally the ECU itself. Voltage checks through the system wiring harness plugs form the basis of most of the electrical checks, and of course the better Fiat/Lancia dealers will be equipped with plug-in diagnostic equipment which pinpoints faults quickly and easily.



10/46: Croma ie. This layout is similar to the Delta turbo 1600 ie, 8v Integrale, Tempra 2l ie in that it utilizes fully mapped injection system. Load is monitored via throttle potentiometer and rpm sensor in conjunction with manifold vacuum (and pressure in case of turbos) sensor. Ignition and injection are controlled by ECU and can only be altered by electronic remapping. A modest increase (around ¼ bar) in fuel pressure will compensate for head porting/blueprinting, but anything more radical requires extensive dyno/computer work to increase injector duration. Note that this illustration shows late (reversed-port) layout with crank-driven oil pump and no auxiliary driveshaft. Moving distributor does not alter ignition static timing – this must be done by repositioning sensor at front pulley end. Tempra also has balance shafts à la Integrale. Note revised water pump and late tensioner. Late cam pulleys are not interchangeable with early type – different offset.



10/47: Weber Alpha schematic layout.



10/48: Parallel fuel injection/spark ignition.

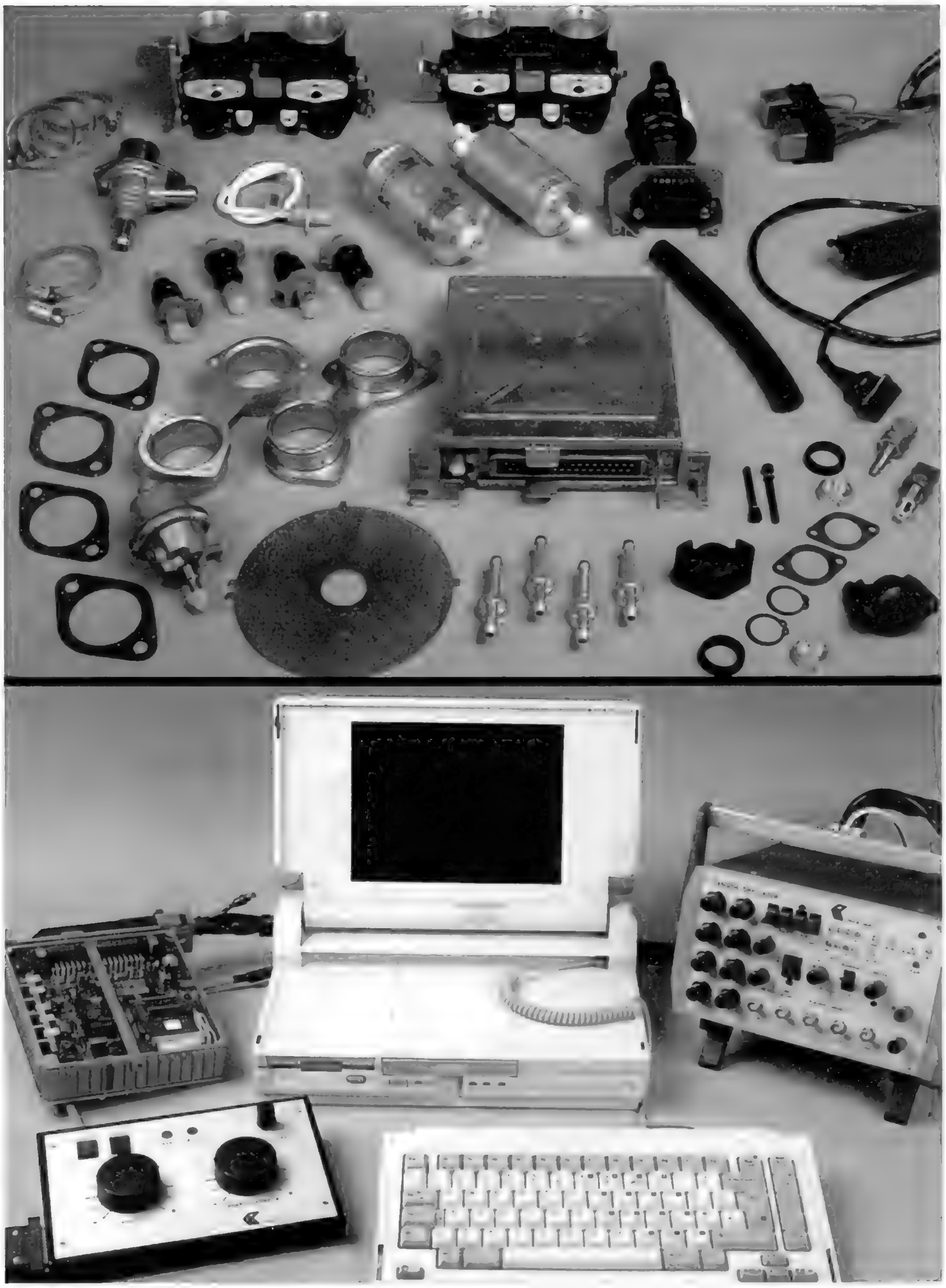
Weber programmable injection ignition (10/47)

This aftermarket system was developed by Weber UK in collaboration with Weber in Italy as an alternative to the conventional carburettor/distributor set-up commonly used. Double throttle bodies (interchangeable with DCOE/DHLA carbs) with 40 or 45mm dia bores are available. This system requires programming on a rolling road or dyno with the use of a portable PC (such as 640 KB Compaq SLT 286 with 20mB hard disc and 3.5in floppy disc for collation of back-up data).

The function of the monitors, ECU and active mapping controller/PC are shown in the layout diagram (10/48).

The Weber system has no airflow sensor (unlike the L Jetronic) or manifold pressure sensor. The throttle potentiometer and engine speed sensors are used to estimate the mass airflow (and hence fuel requirement) from a calibration map. The system is mapped by selecting spark advance and injection tune at various positions of throttle angle and engine speed, *(see Case History No 5)*. All the variables are stored on disc and later transferred to the EPROM chip. An additional function is that the programme can be edited before transfer to the EPROM ('blowing' the chip) by running the engine in the car with the portable computer connected to the ECU so that in-car adjustments can be made.

Due to the precise fuel calibration, better atomization and optimized ignition



10/49: Weber programmable system hardware!

curve, primary advantages of the system are:

1 Torque increase of up to 5%, power increase 10%-plus over carbs.
2 Greatly improved tractability at low speeds.
3 No distributor is required – plugs each have separate coils.
4 No recalibration is needed once set up.
5 Reduced risk of detonation

Once the load-testing calibration has been carried out, corrections for warm-up enrichment, sustained full-load running, snap acceleration and cranking enrichment can be added to the programme. The system even allows for battery voltage variation!

Note – CAUTION: For selection purposes, the 40mm throttle body is roughly equivalent to a 45 sidedraught carb (40 choke) and the 45 throttle body to a 48 carb (42 choke). 'Bolting on' this injection requires the same careful attention to matching inlet cfm/velocity characteristics to the engine specification as with a carburettor installation.

PART THREE: AIR INTAKE COMPONENTS

Filtration

An often neglected topic, good air filtration, is important to sustain engine power (and possibly enhance it compared with a paper element) in dusty conditions, to minimize engine wear due to inhaled particles, and prevent power loss due to adverse inlet tract wave effects.

The greater part of airborne dust is formed by quartz particles, the remainder being mainly vegetable components, and a good race filter will trap particles down to around the 6 micron (μm) range. Inhalation of dust particles, particularly when mixed with rubber from race tyres, can have a devastating effect on the life of the cylinder bores, rings and valves/seats.

Foam filters (notably ITG, Pipercross) have been widely adopted in motorsport (from Formula One down) because of the all-round filtration they offer compared with contemporary filters (gauze/paper) equipped with a steel cover, and because of the exceptional laden airflow characteristic of the foam types used (with their appropriate wetting oils). Inlet tract ram-effect is disrupted when a steel cover is placed over the carburettor/injection intake mouth unless a large standoff (at least 4") is used. The flame retardant effect of foam filters is quite good (unless they are soaked in fuel!). **(10/50)**

Always leave at least 1" between the intake mouth and inside of the filter assembly. A one-piece backplate is ideal for twin carbs because it holds the carbs in balance. Select the filter in conjunction with the *manufacturers* to ensure adequate capacity. In extremely dusty conditions, a dual filter system can be employed, *ie* a standard filter with an 'over filter' on top.

Rampipes

Adding length to the outside of the carbs or throttle bodies (Weber programmable) can increase ram-effect – and by optimizing the inlet tract length relative to the wave effects, lead to enhanced torque if the rampipe design is good. Rampipes have the distinct advantage that, whilst lengthening the inlet tract to suit the negative wave frequency, added rampipe length will not cause the fuel to drop out of circulation – unlike an excessively long inlet manifold. (An increase of rampipe length from 2" to 4" can produce a comfortable increase of 2lbf ft of torque mid-range – although short pipes give better top-end – on a 2*l* TC.) The Pipercross rampipes shown are

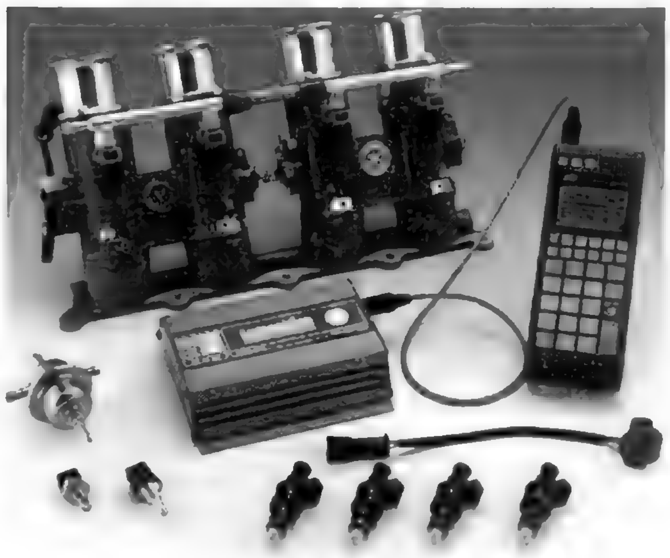
10/50: Pipercross one-piece backplate foam filter and shallow 'roll back radius' alloy rampipes. Those shown are fitted to Lumenition FI and DHLAs. Cheap horsepower!

an outstanding example of design effectiveness (maximum roll-back radius – sharp edges presented to the airstream create turbulence and reduce flow, and the narrow included angle keeps the airflow laminar) and low cost – no serious competition TC should be without them! These pipes are designed to fit neatly inside the Pipercross range of filters and secure the backplate(s) in place. Lengths are available from Pipercross (or ITG) to suit the demands of the particular event.

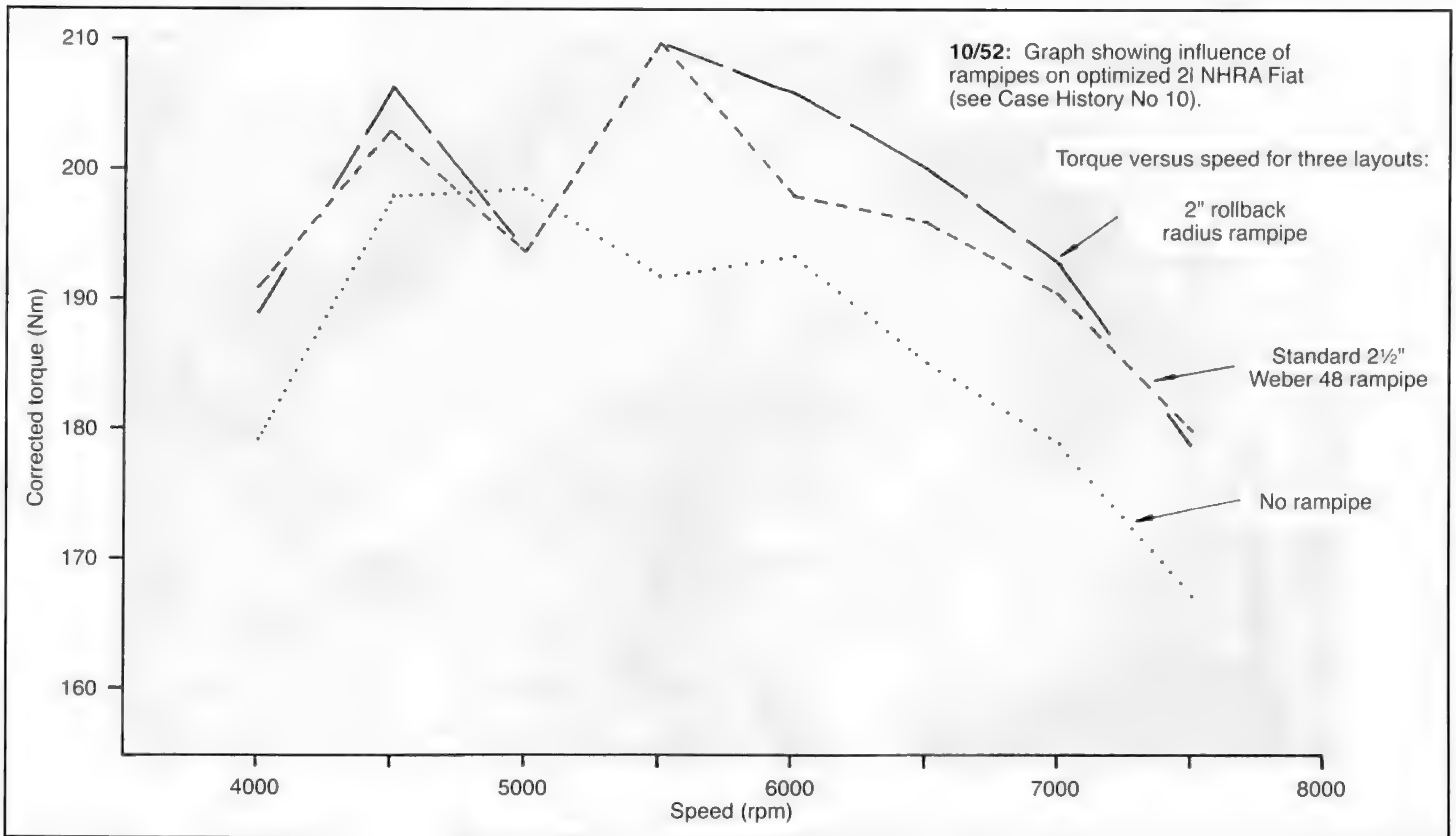
Cold air intake

By use of appropriate ducting (it should ideally fully enclose the filter) or an airbox around the intake mouth combined with ducting to a remote filter, it is possible to feed cooler intake air to the fuel system than is the case when the intake air is merely drawn from the engine bay. The power loss (and risk of detonation) from hot intake air should not be underestimated.

Always ensure that the air intake is shielded from hot air drawn through the radiator/oil cooler. Additionally, a useful amount of over-pressure can be generated in the inlet tract by siting the air intake duct in a suitable part of the vehicle. The induction ram from a vehicle travelling at over 80mph is well worth capitalizing on. (Note that the jetting of race carburettors must be matched to the air intake temperature. Running the cylinder head too hot significantly reduces power by lowering the charge density in the ports and inlet manifold – *see Case History No 10.*)



10/51: Lumenition mapped fuel injection/ignition system is calibrated using hand-held PC shown.



10/52: Graph showing influence of rampipes on optimized 2l NHRA Fiat (see Case History No 10).

HOW DOES A CHIP WORK?

by Gerard Sauer (reprinted by kind permission of *Cars & Car Conversions* magazine)

The function of a microchip can be roughly compared with that of an instruction manual containing a set of instructions which allow you to correctly operate a complicated machine. For example, on page 27 this manual may tell you that at 2500rpm you should switch a lever from (a) to (b) to increase the cooling, in order to prevent overheating. On page 1, it might have told you what to do to start the machine, while on page 35 you can find out what to do if the maximum rpm is approached, and so on. Provided you keep reading the instructions and keep doing as the book says, you should be able to operate the machine very well and within its design limits.

In a similar way, the microchip contains lots of instructions which the computer part of your ECU keeps scanning through at phenomenal speed (a rate of approximately a million instructions per second) and then compares the inputs that it reads from the sensors around the engine, such as: engine rpm, coolant temperature, manifold pressure and so on – with the relevant values stored in its memory. This, in turn, makes it possible for the calculating part of the ECU to select and output the correct commands/voltages to a series of actuators such as the injectors, the ignition amplifier and coil.

There are some operational details that are important to understand. The most relevant of these is the fact that the sensors and actuators both work with analogue data, usually electrical signals in the 0–5 volt range, while the processor in the computer will only recognize digital inputs and outputs. As a result, analogue-to-digital converters and digital-to-analogue converters are needed to translate and make suitable the signals coming in and going out. This allows the ECU to calculate and react very quickly, something which it would not be capable of if it had to work with real-time analogue signals: it separates the calculating and interpreting functions.

Another fact relevant to this fuel-injection and ignition-control environment is that the ECU can work with a great many inputs and outputs at the same time. This makes it possible to consistently run the engine closer to its performance maximum reliably.

Referring back to the instruction manual analogy, the company modifying a turbo engine chip, for example, would – metaphorically speaking – look up the page for boost pressure and, where it says that you should open the valve that is in-line with the wastegate actuator when the boost pressure values approach 14lbf/in², they would replace this with, say, the number 18 instead. This would then leave the valve closed until the number 18 – or 18lbf/in² boost – is registered on the constantly-monitored pressure gauge. In some cases, chip tuners have simply removed any reference whatsoever to a limit, thereby making the 'instruction manual reader' think that there is no such limit to worry about.

Obviously, this is a simplistic representation and is not exactly what happens inside an ECU, but it summarizes the idea. In practice, the 'instruction manual' is written in a language suitable for the machine, and not the operator, and therefore the pages we mentioned are not in any easily recognizable format or order. In addition, the sheer volume of information stored would mean that you need to work your way through hundreds of pages of information. Finally, even if you did understand all the language, and knew where all the functions and instructions were located, you would still have to work out what actions and effects are related to each other – or not, as the case may be – in order to make changes that won't destroy the engine or the turbo.

Most chip tuners need to learn about the program that they intend to change, and have become very adept at unscrambling this data, usually with the help of a device called an emulator. By putting the ECU through its paces in a simulated 'dry' run, they can identify certain oft-repeated program actions, and so identify the various pages and instructions that they must modify to raise the boost pressure or change the opening times of the injectors.

Manufacturers spend months and months defining and refining all the necessary parameters of a car's standard ECU, so naturally it would be impossible and totally uneconomic for the chip tuning company to reprogram the entire ECU. And unnecessary, since many of the instructions can remain as they are, even after modifications have been carried out. It is crucial that the new instructions programmed into the ECU by the tuners, via a new chip, are complete and appropriate, including new limits and fuelling numbers which won't damage the engine.

That this has not always been so in the past was amply demonstrated when a particular chip program from a Lancia Integrale 8v was examined and compared with the standard item. From an investigation of the relevant figure it was clear that there was just one changed number. This was the number that determined the maximum boost value. By removing the maximum limit in such a simple way, there was no telling when next the engine would run short of fuel and detonate to destruction, probably before the driver even realized that anything was wrong at all. What was even more galling was the fact that this change could be done with a suitable programmer in a matter of seconds, to 20 or so chips at once.

With the chip hardware itself costing around £3, and with modified chips selling for anything up to £350, it is not hard to see the enormous profit potential. When they realized how easily it could be done, many clever conventional tuners then enlisted the help of so-called computer hackers to do all sorts of things to the ECUs, and naturally-aspirated cars began to receive attention, too. Speaking to several people who had experienced these early naturally-aspirated 'tuned' chips unearthed some truly horrendous stories.

So, what can be done to protect the engine, and how do you separate fact and fiction? The answer is to ask a few key questions before purchasing a chip and insisting on all the answers in writing. Before buying, get some answers to the questions below. Clearly, there are lots of things to consider when you are going to submit your car to the dark art of reprogramming. However, it is equally valid to point out that, if done correctly, the results can be good and worthwhile. The difference lies in the care that the company takes in doing it and what strategy of change they have followed

Questions to ask before deciding on a particular modified chip:

Turbo cars

1 Has the tuning company set new limits of maximum boost and, if so, at what level?

Answer: The maximum boost limit is defined by a number of factors, but the main ones are: capacity of the injectors to supply adequate fuel to prevent excessively lean mixtures; head gasket clamping limits; elastic limits of the

bottom end; and turbo thrust bearing limits. There must be a new clearly-defined limit of boost and this should be accompanied by an over-boost fuel/ignition cut-off if exceeded.

2 How is this overboost limit affected: is it an ignition cut-off, a fuel cut-off, or both?
Answer: The ignition should cut out first, then the fuel.

3 How much reprogramming have they done?
Answer: The better the programmer the more so-called 'sites' will have been altered to suit the new limits. If different injectors are part of the package, then almost all the sites will have had to be changed. Also, things like cold-start and hot-soak areas should have been dealt with both on the ignition and on the fuelling sides.

4 If the injectors have been changed, are they not over-fuelling at low engine speed?
Answer: The manufacturer will always specify an injector with a dynamic range that gives the best atomization at low load/rpm. That means that any increase in injector size to cope with the additional fuelling requirements at high boost/rpm usually affects this area of operation. The simple solution is to attempt to modify the engine without having to change the injectors. If they have to be changed, though, a dual set is usually the better solution since for off-boost performance the first set can be kept in operation as standard, and the second set can be kept fairly small as well. Naturally, this is more expensive.

Normally aspirated cars

1 Has the tuning company set a new rpm limit and what is it?
Answer: It is most important that if the rpm limit has been changed a sensible new limit has been set; the fact is that the standard valve/spring combination won't allow greatly increased revs anyway on most cars. The rest of the rotating components also do not like very high revs as they have usually been designed with the standard rev limit in mind. A target for a new limit is usually between plus 300–500rpm.

2 Are the mechanical components of the engine capable of enduring the increased stress?
Answer: The facts are that there will always be spare capacity designed into the engine's components to avoid early or unexpected failures (or, more accurately, warranty claims!). It is the level of increase that is going to determine the new loads and stress patterns for the engine. Sensible tuners will tell you what to expect, but remember that careful maintenance and preparation also has a lot to do with it. Too much advance and the engine will detonate, and no amount of strength in the standard unit is going to prevent failure if that happens.

3 How much horsepower has been gained, if any?
Answer: This is always the difficult one to answer for most tuners. The power gained is in relation to what was there before. The switched-on tuner will test your engine beforehand and check for cylinder leakdown and compressions before attempting to tune anything. On normally-aspirated engines increases are hardly possible without an increase in revs, usually in concert with other changes such as air filters, exhaust manifolds and suchlike. A good target number that is fairly realistic seems to be gains of around 5% in the mid-range/part-throttle area and some small gains just below that. The best you can hope for is 10%: beware, the rest is just 'pub talk'.

4 Have they modified the mid-range part-throttle sites?
Answer: Most gains are to be found here since the manufacturers tune their engines, including the ECU, to give good results in the statutory Government fuel consumption tests, so that the marketing department can shout about the fuel economy of the car. Once these tests have been completed the manufacturer must make all the cars to this specification by law.

5 Is there any chance of the system over-fuelling?
Answer: The answer should be negative, but you should be aware that this is difficult to prove except on a rolling-road. In case your fuel consumption goes up dramatically, make sure the tuner agrees to show you on the rolling-road with an infra-red gas analyzer that there is no excess fuel being injected. Hot engine values should not exceed 12.5:1 air-to-fuel ratio.

All conversions

1 How much testing and reliability work was done on this conversion?
Answer: To be certain that the increased power is not going to ruin the rest of your drivetrain and that the brakes are still adequate, the company should test the car for these problems over a fair distance and under controlled conditions. It is not good enough to say that they have not had any previous complaints: some people simply don't complain, and in any case they may not use the car as hard as you are planning to. A guideline distance for testing is 1500 miles of sustained hard work, with cold-start and hot-start tests, together with a thorough engine, driveline and brake check afterwards.

2 What is their advice on the modified engine's life expectancy?
Answer: No matter what you do, increased-performance modifications are always going to reduce engine life to some extent if your maintenance plan doesn't take account of the changes but, with modest increases in boost/power, they should advise you on much more frequent oil changes and maintenance checks. Anything more than modest increases in boost pressure (4–6lbf/in^2) or engine speed (3–500rpm) and the engine life will be shortened.

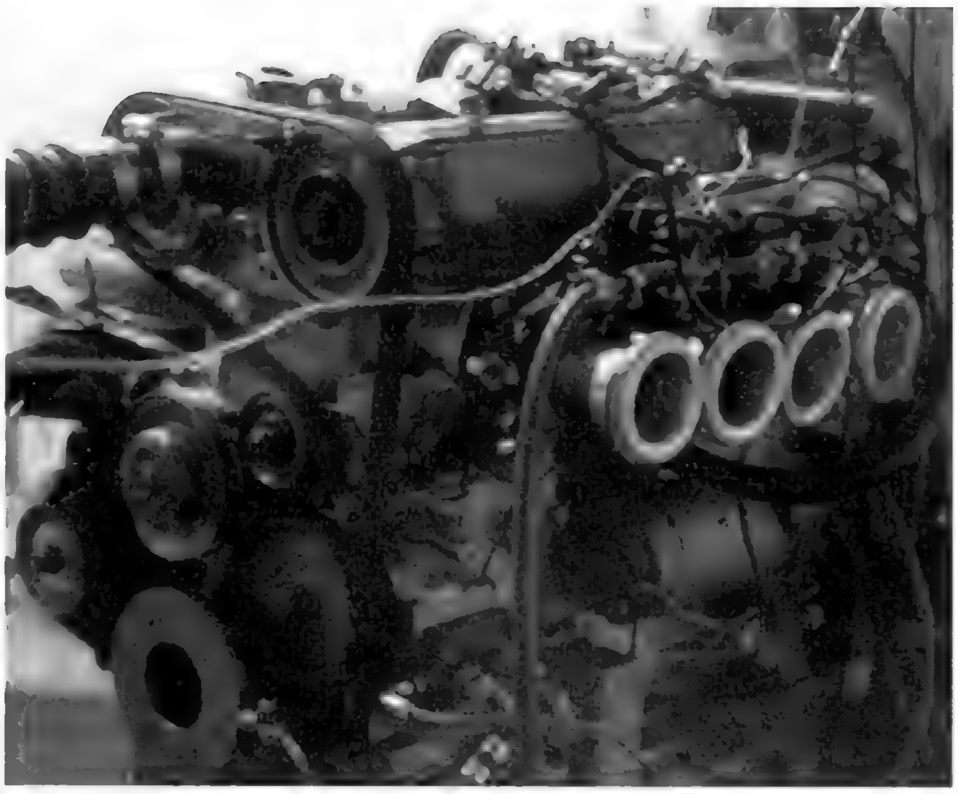
3 Have the modifications been emissions-tested?
Answer: With new legislation coming along that will affect all cars, whether modified or not, this is increasingly important. Catalytic converters hate over-fuelling, and stop working altogether soon after. This could be a nasty surprise when you present your pride and joy for its first MoT.

[*Author's note:* This is one of the most informative and concise summaries of the subject I have seen. It confirms my opinion of Gerard Sauer as *the authority* on the subject.]

CASE HISTORY No 5

Owner	Colin Haggett
Engine No	203
Type	Fiat 2049cc (85mm bore) St III
Use	Rally
Tested	Warrior, Aug '91
Rig	Superflow

Specification:
GC St III 'Tarmac' cams.
45mm throttle bodies.
46/40 race valves.
Triple springs.
Forged pistons 11:1 CR.
Steel (ultra-light) flywheel.
Weber-Marelli fully mapped ignition/injection.



CH5/1: Note chopper on front pulley which allows sensor to measure rpm and TDC position. GC oil take-off plate allows ready connection of oil heat exchanger and oil pressure gauge. Spun rampipes with generous roll-back radius reduce airflow (cfm) entry losses.

In the flexibility test, this engine would pull full-throttle from 2000rpm and developed 49lbf ft torque! A testament to the effectiveness of the injection/ignition system – with 320° cams! Extensive tests had to be carried out using different throttle plate angles at various speeds; the ignition settings were optimized at each position. For example, the engine, primarily due to the large intrusion of the piston dome into the combustion chamber – interfering with the flame-front – required 44½° ignition advance at 8000rpm at 87% throttle!

The optimum result is recorded as curve B on the graph. The engine has a broad spread of torque from 5000rpm (139lbf ft) to 7150rpm (140lbf ft) with a peak result of 149lbf ft, which should result in a tractable unit for rallying.

This engine was tested with the same 4-1 22" manifold as Tom Casey's Hot Rod engine. Torque would almost certainly have been improved by at least 3–5% had a 36" 4-1 been used.

The ignition timing results are recorded on the same graph and are particularly interesting. Note that the curves differ widely from the standard 131 2*l* centrifugal (non-vacuum) advance curve, except that around 5500rpm the simple standard distributor produces about the right amount of advance. The mapped system demonstrated how effectively it can enhance the torque curve; clearly no conventional unit could do this. Note that when torque is good at full-throttle the advance requirement tends to be low and *vice versa*.

The fuelling requirement showed that the engine needed a very rich mixture at full-throttle, but the facility of dialling in the requisite ignition timing ensured that it burned cleanly. The fuel delivery had to be raised 40% at 8000rpm, between 73° and 90° throttle, which emphasizes what can happen to fuel consumption with a heavy-footed driver!

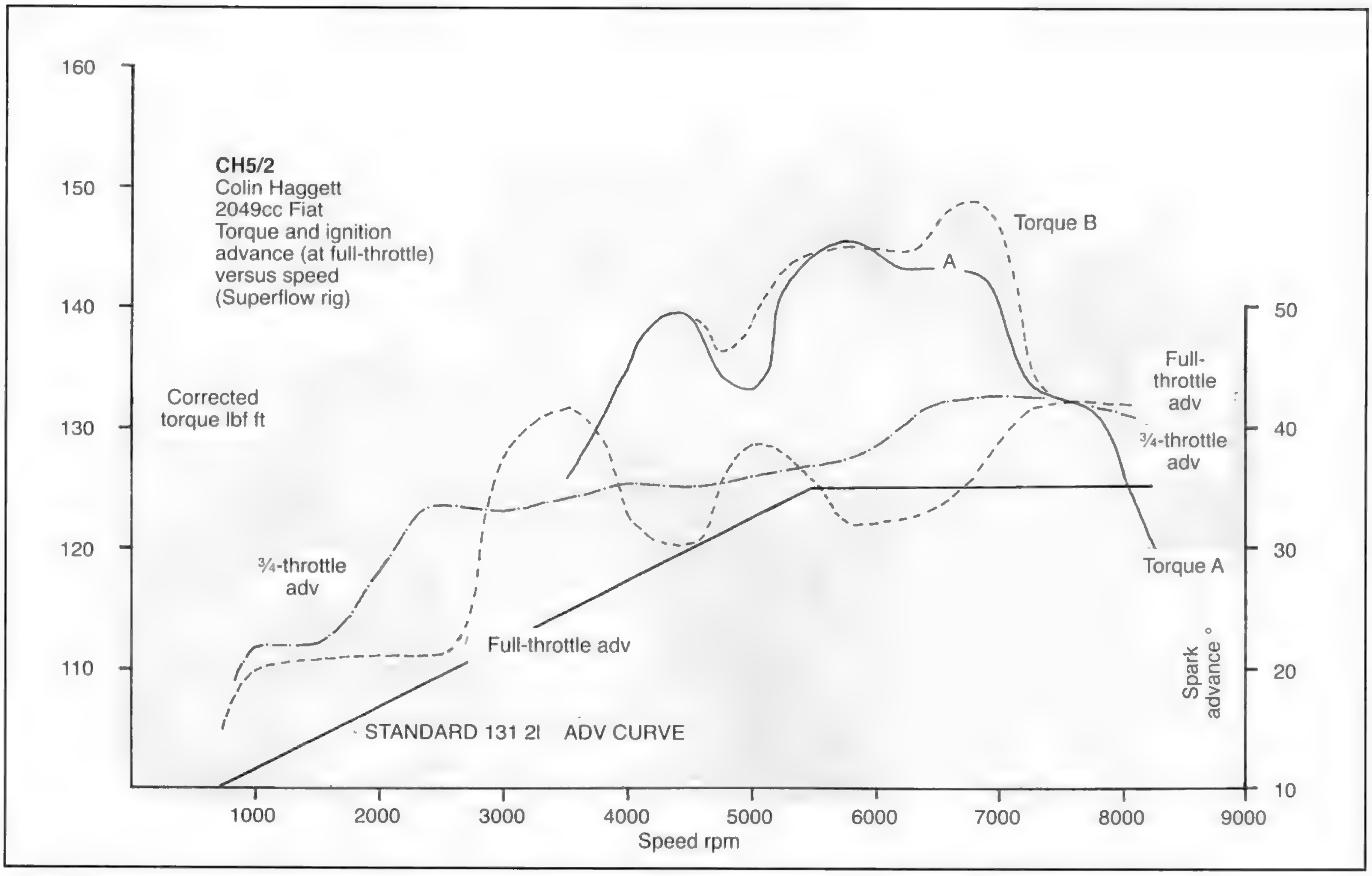
The full list of ignition timing *versus* engine speed *versus* throttle plate position is reproduced in the accompanying table.

ALPHA ENGINE MANAGEMENT CALIBRATION TABLE : ***Case History No 5*** **– August 8, 1994**

SPARK ADVANCE MAP: FUNCTION OF THROTTLE POSITION AND ENGINE SPEED

Throttle position (degrees) / Engine speed (rpm)	719	1027	1506	2019	2499	3012	3525	4005	4518	5031	5750	6503	7256	8009	8762
0.0	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.50	10.50	10.00
5.0	28.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00
15.0	26.00	20.00	32.00	36.75	39.25	36.50	36.75	37.25	38.00	36.50	35.00	35.00	35.00	33.50	32.00
23.5	15.00	18.50	30.00	34.25	38.00	38.00	38.00	38.00	38.00	36.50	35.00	35.00	35.00	33.75	32.00
27.0	12.50	13.00	33.00	35.00	37.00	38.50	38.00	38.00	38.00	37.00	35.00	35.00	35.00	33.75	32.00
32.0	14.00	14.00	28.00	31.50	36.50	39.50	39.75	38.25	38.00	36.00	35.00	35.00	35.00	33.50	32.00
38.0	15.00	18.00	22.00	22.75	33.00	41.25	41.25	41.75	41.25	36.00	35.00	35.00	35.00	32.75	30.00
43.0	15.00	18.00	19.50	25.25	29.50	44.25	43.25	42.25	42.75	40.75	39.75	39.75	39.75	37.50	34.75
48.0	14.50	13.75	14.00	26.00	27.00	35.00	34.00	34.75	34.00	32.50	31.00	31.00	31.00	29.25	26.00
54.0	10.00	10.00	10.00	26.75	26.25	29.25	29.50	34.25	34.00	39.00	38.25	38.75	38.00	37.00	37.00
59.0	15.00	22.00	22.00	27.00	27.00	29.25	33.25	37.00	35.00	37.25	38.00	41.00	43.00	43.00	40.75
63.0	15.00	22.00	22.00	28.00	28.50	29.50	35.00	35.50	38.00	36.00	36.00	38.00	42.00	44.00	43.00
67.0	15.00	22.00	22.00	28.00	33.50	33.00	34.00	35.25	35.00	36.00	37.00	42.00	42.25	41.00	38.50
73.0	15.00	20.00	22.00	30.00	36.25	35.00	33.00	31.00	32.00	35.50	36.75	36.25	42.50	41.25	44.50
78.0	15.00	20.00	22.00	28.00	28.00	36.00	37.00	32.25	33.00	37.00	34.00	36.00	41.00	43.00	43.00
90.0	15.00	20.00	21.00	21.25	21.25	37.75	41.75	32.50	30.00	38.75	32.25	33.50	41.75	42.00	42.00

MAX ADVANCE REQUIREMENTS



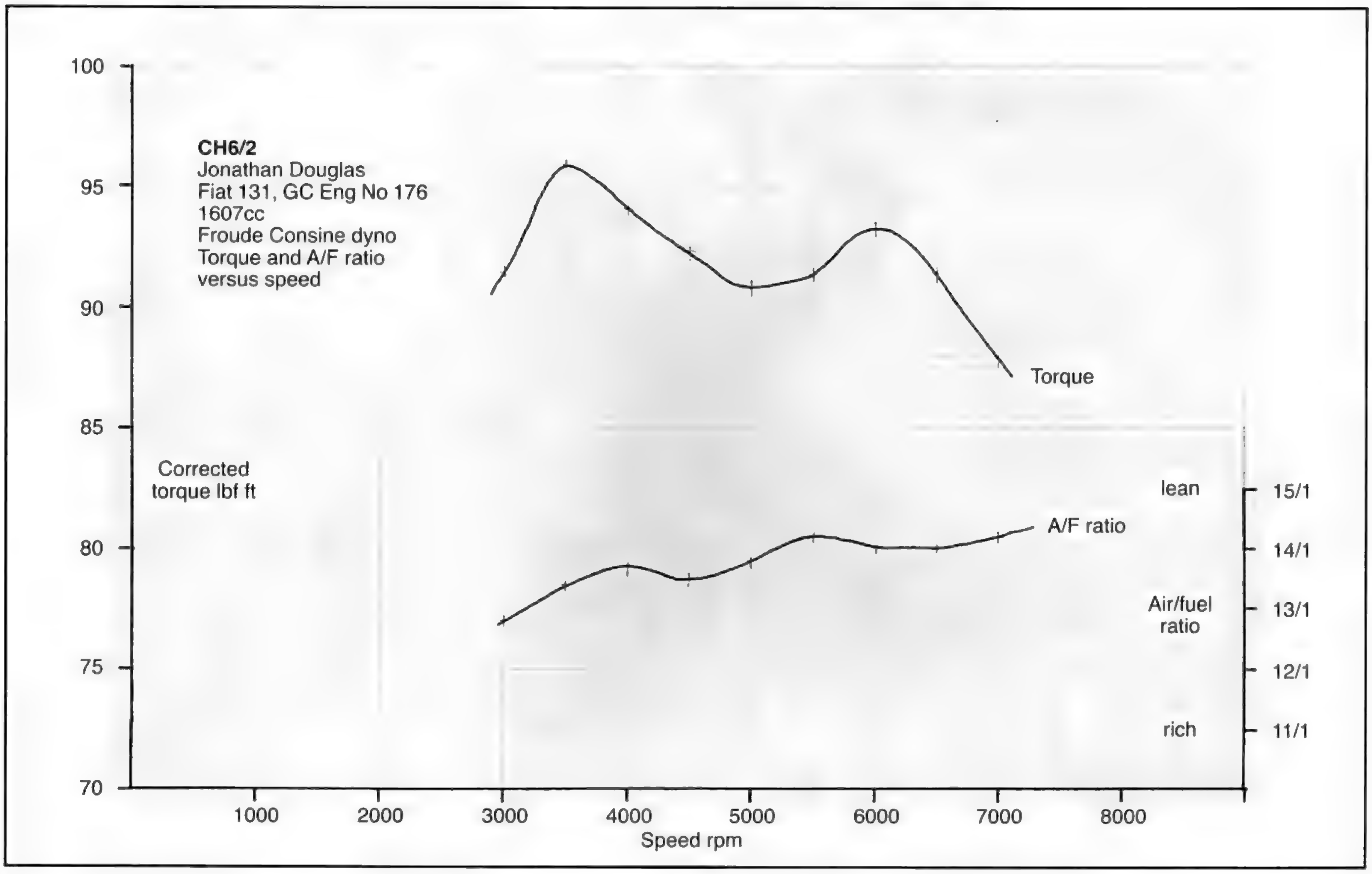
CASE HISTORY No 6

Owner	Jonathan Douglas (Director Induction Technology Group Ltd – air filter manufacturers)
Engine No	176
Type	St II Fiat 131 1600
Use	Road-race Morgan
Tested	JE Engineering, Coventry, May '93
Rig	Froude Consine

Specification:
84.6 bore, 1607cc.
Partially ported/blueprinted seats.
Standard 42/36 valves.
Standard 131 2*l* carb and manifold.
10:1 CR forged pistons, lightened balanced flywheel/crank assembly.
28/68 68/28, 10.6mm lift cams, dual interference springs.

This engine was designed to race in the Morgan Challenge and as such was required to run with a single down-draught twin-choke carburettor. The engine was dyno tested with a 4-1 exhaust manifold. It was found that below 3000rpm the engine would not hold full-throttle and there was significant fuel stand-off at

CH6/1: Jonathan Douglas leading (and about to beat) three Plus-8 Morgans and a Supersports TR4-engined Plus-4 at Silverstone with his well-balanced Fiat 131-powered car. (Photo Fred Scatley)



the carb mouth. Unfortunately, it was not possible to test the engine above 7000rpm because the dyno performance envelope did not satisfactorily accept loads below 80lbf ft. Maximum bhp was 117.2 (corrected) at 7000rpm.

It proved difficult to enrich the mixture at high revs with the single carb, a problem which was later to cause difficulties with engine No 204 *(see Case History No 4)*. However, the high air velocity generated by the competition cams led to a reasonably strong torque curve, peak torque at 3500rpm (similar to standard engine) and a satisfying 'hump' in the torque curve at 6000rpm. In competition, the car proved very competitive, beating not only other 1600-engined cars but also several V8s.

Final jetting was: idle 50/50, air corrector 140/115, mains 135/132, chokes 26.5/28, enrichment jet 85.

Jonathan Douglas wrote (on 16/11/94): 'The only seriously non-standard part of my car is the engine... The modifed +8s are extremely quick in a straight line and lap about 15–20% quicker than my near-standard 4/4 1600. The most important thing about the Fiat engine in a Morgan is that it is light enough to give a well-balanced and fast-cornering car; cars that have been built with more highly tuned 2-litre engines lap as fast as all but the most highly modified V8s, even though they are giving away 80–100bhp, simply because their cornering is so much faster and more balanced.

Some lap times for the 1600: Mallory Park 0.58.9; Silvestone (Club '93) 1.20.4; Pembury 1.14.7; Brands Hatch (Indy) 1.0.6. With the old 2-litre engine (standard cams and valves, gasflowed head, 1600-type (high-CR) pistons, standard carb) Silverstone was covered in 1.17.6. That also gave about 125bhp, but better torque...'

CHAPTER 11

IGNITION SYSTEMS

Including Digiplex and Microplex

The purpose of the ignition system being to initiate combustion, it is vital that the system produces an electrical discharge (spark) of the necessary intensity at the right place in the firing cycle. If the spark is too weak, particularly when the mixture is rich or the cylinder pressure is high, the engine will not run. If the spark occurs in the wrong place, at best, power will be lost, and at worst, engine damage may occur.

To create a spark (which is the means of initiation of the explosion of the fuel/air mixture) requires a potential (voltage) difference to exist between the positive (centre) electrode of the spark plug and the negative, earth electrode, *ie* the body of the plug. The contact-breaker and more advanced modern systems all employ a transformer coil to generate this discharge at the plug. The primary and secondary coils are wound concentrically around an iron core, and when the current to the primary is switched off, the collapsing magnetic field causes a current to be generated in the secondary windings. The means of switching can be either by a simple contact-breaker, which is actuated at a speed to match the ignition cycle (half engine speed for the four-stroke TC), or by a transistorized system (11/1).

The contact-breaker closes to charge up the primary windings of the coil; and this phase is known as the 'dwell' period (measured in degrees or percent). When it opens, arcing tends to occur across the contacts as the current in the primary winding opposes this change of condition and a capacitor (condenser) is connected across the points to absorb this current. When the points close again, the capacitor discharges into the coil, so helping to build up the primary current. It is a fact that a greater current is generated by collapsing the primary magnetic field than creating it, thus it should be remembered, when 'statically' timing a TC, that the spark occurs when the contact-breaker *opens*.

The problems with a contact-breaker system are that the system is operated at full battery voltage, using a large current drawn from the main starting system; that the performance of the points at high engine speed is degraded by the inability of the contact-breaker spring to control the system accurately. The points themselves are subject to wear, and since inevitably some minor arcing occurs across them, even on a well-maintained system, the points tend to 'grow', closing up in the process and affecting the dwell period/timing.

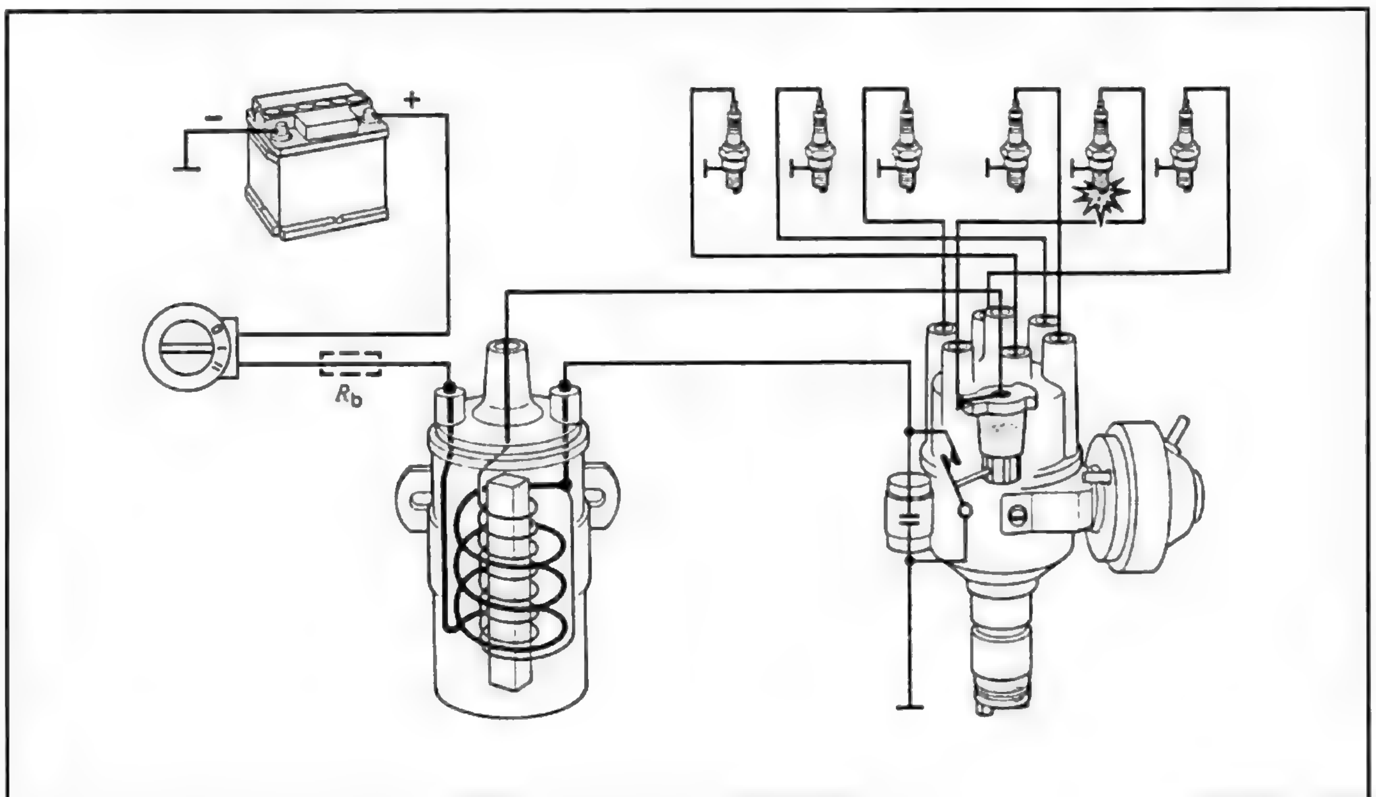
Transistorized systems overcame these pitfalls; a very low switching voltage/current is required, almost zero maintenance, and the intensity and duration of the spark is greater.

The earlier TCs used a contact-breaker system; electronic ignition (transistorized) was introduced on the Fiat 132 2*l* and 131 versions, plus certain Lancia models in the mid-70s. The electronic ignitions were basically of two types. In the Bosch system, a pressed-steel four-blade reluctor rotates around an inner coil to generate the switching current; as each of the reluctor arms passes the pickup in the distributor the current is fed to the amplifier unit. With the Marelli system, the reluctor is a steel four-lobe rotor, but it generates the current in a similar way.

Dwell variations can occur on a CB distributor due to worn bearings, especially if it is of the type with the centrifugal weights mounted above the CB points baseplate. Similarly, the performance of the points themselves can cause the dwell to alter at high rpm. It is better, therefore, to examine the dwell angle with a meter and ensure that at both high and low rpm it falls within the stated data, than to set them with a feeler gauge and 'hope for the best'. Use of electronic ignition eliminates dwell variation.

Useful data to CB systems

Dwell angle		52–58 degrees
CB points gap		0.016"–0.018"
Coil resistance		
	primary	Marelli 3.14Ω ±0.13
		Bosch 2.6–3.1Ω
	secondary	Marelli 9400Ω ±400
		Bosch 8500–12000Ω

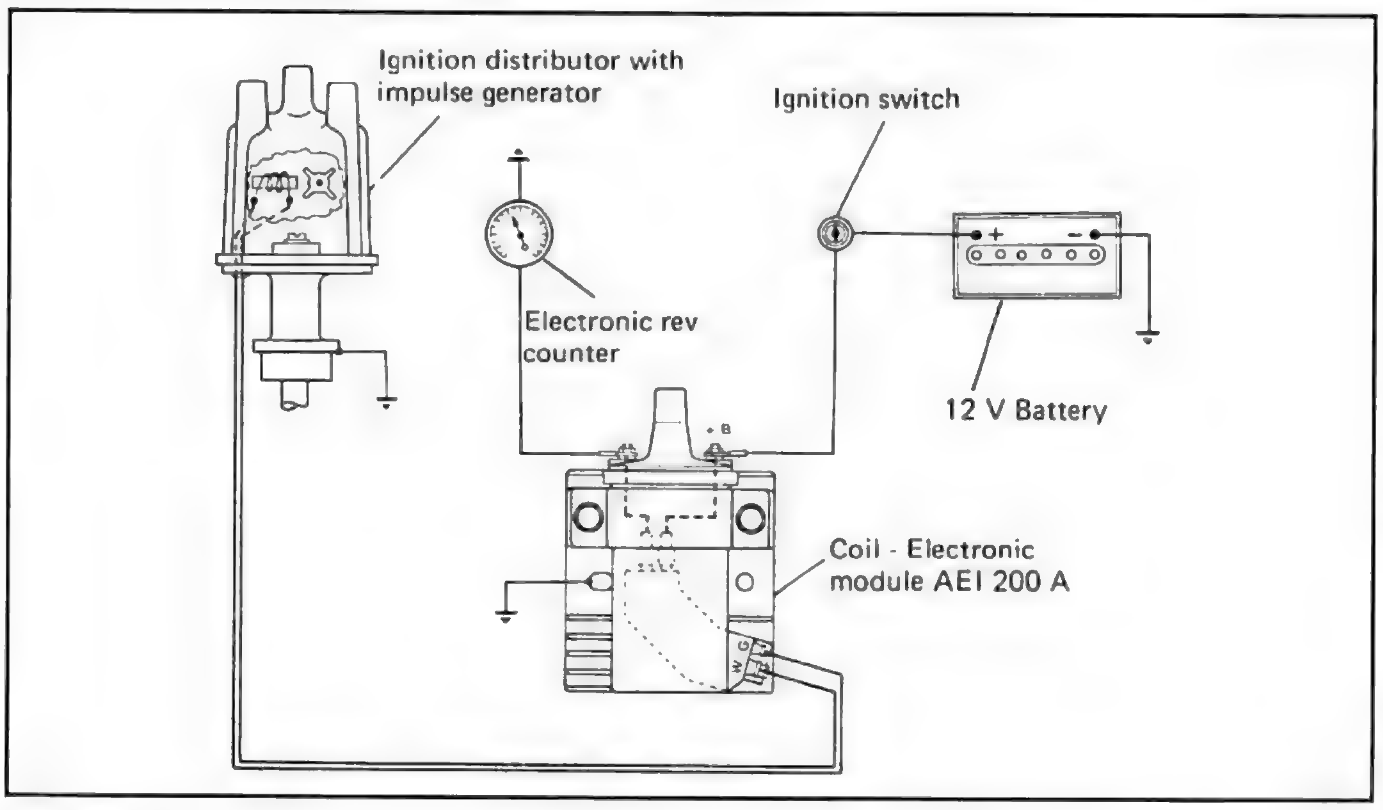


11/1: Diagrammatic layout of ignition coil and contact breaker system.

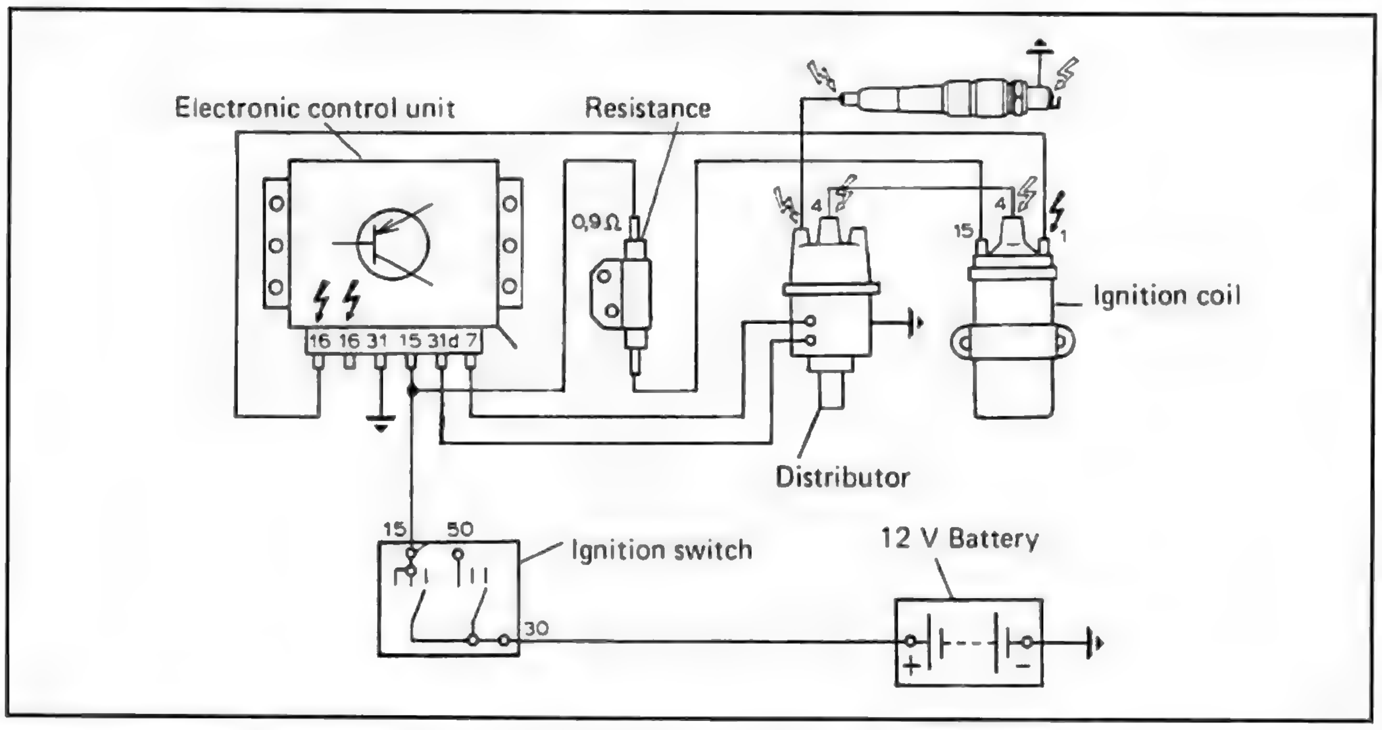
	Marelli	Bosch
Inductive discharge electronic ignition	Marelli AEI 200 A	Bosch 0.227.100.014
Firing order	1 – 3 – 4 – 2	
DISTRIBUTOR		
Make	Marelli	Bosch
Type	SM800 AX SM 801 AX	0.237.001.002
Distance between 4-pole rotor and stator pole	0.30 ÷ 0.40mm	—
Ignition coil winding resistance of electromagnetic impulse at 23°C	730 ± 7 Ω	1100 ± 10 Ω
RESISTOR		
Make	—	Bosch
Type	—	0.227.900.002
Resistance	—	0.9 ± 0.05 Ω
IGNITION COIL		
Make	Marelli	Bosch
Type	BAE 207 A	0.221.122.012
Ohmic resistance of primary winding at 20°C	0.75 ± 0.81 Ω	1.2 ± 1.6 Ω
Ohmic resistance of secondary winding at 20°C	9500 ÷ 11 500 Ω	6000 ÷ 10 000 Ω

11/2: Electronic ignition technical data for Fiat 131.

Bosch/Marelli electronic systems (11/3, 11/4)



11/3: Diagram showing Marelli inductive discharge ignition with electronic module. (Fiat Auto SpA – copyright reserved)

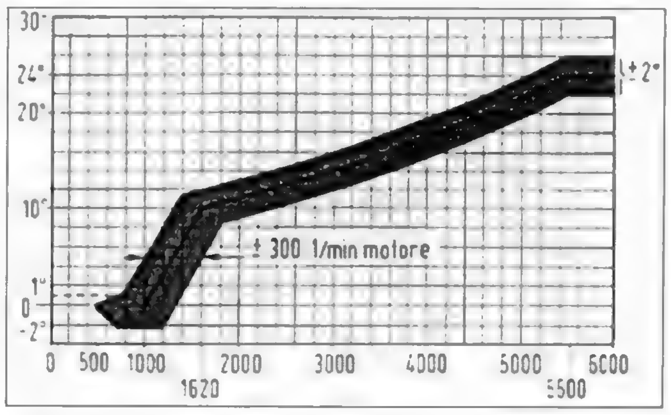


11/4: Diagram showing Bosch breakerless ignition system with electronic control unit. (Fiat Auto SpA – copyright reserved)

Electronic ignition should always be sought if the engine is intended to run over 7500rpm, where 'points bounce' will lead to power loss (as much as 5% at 8000rpm over an electronic system). This, since the availability of aftermarket systems is virtually nil (except for 'mappable' systems), will invariably mean a Bosch/Marelli system from another model.

Ignition timing/advance principles

The combustion phase tends to take place over a fixed time period, whereas as the engine rotates faster, the crank moves a further distance in this time period. Therefore, to ensure maximum 'push' on the piston just after TDC, the engines incorporate systems (either mechanical or electronic) to advance the ignition timing as the revs increase. Prior to the introduction of 'mapped' systems this was achieved by the use of spring-loaded centrifugal weights in the distributors. Centrifugal force causes them to fly outwards as the speed increases, thus swivelling the contact-breaker (or reluctor) baseplate relative to the distributor shaft and advancing the ignition. Their movement is controlled by springs and stops. Thus by matching the weight of the bobweights, spring rate and position of the stops to the ignition requirement of the engine, a characteristic curve reasonably suited to the engine's needs can be produced. All the TCs (with centrifugal distributors) use 'two-stage' advance, and the characteristic curves for end-drive (non-mapped) and block-mounted distributors are broadly similar.

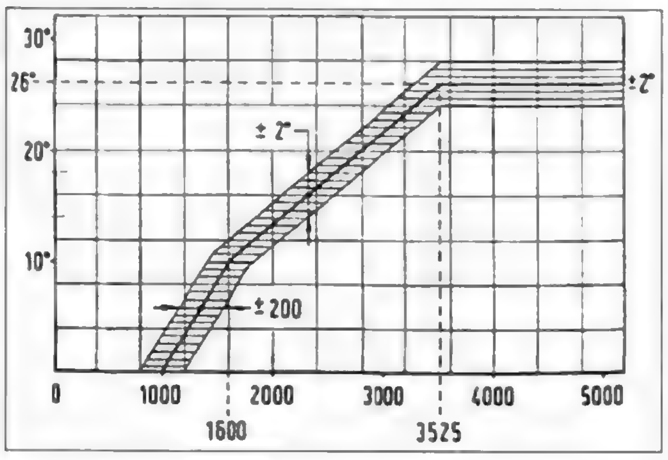


11/5: Diagram showing automatic centrifugal advance of magnetic impulse distributor in 2000 2ACT engine. (Fiat Auto SpA – copyright reserved)

Interestingly, the 124 Abarth Spider (8v) was designed to create maximum advance at lower rpm (11/6).

[*Author's note:* For competition purposes, owners may assume that all the electronic/CB distributors have the same characteristics.]

INITIAL FAULT FINDING WITH SYSTEMS ON PREVIOUS PAGE		
SYMPTOM	POSSIBLE CAUSE	REMEDY
CB POINTS SYSTEM		
Engine hard to start, misfire especially when cold, engine idles erratically	Weak spark	Check battery with heavy discharge tester Check volt drop across points; when closed should be zero Check condition of points/coil; they should be regularly inspected on a competition engine. On a road car replace points every 6000 miles or at least every 12 months. If points OK, suspect/replace condenser
Engine lacks power/'pinks'	Points closed up	Check/replace and reset
ELECTRONIC SYSTEMS		
Engine hard to start, misfires	Faulty amplifier	Replace (Bosch units are only available from manufacturer, Marelli amp units are available from Unipart)
Note: Always check the connections on electronic systems before replacing components.		



11/6: Advance curve for Fiat Abarth 124 Rally. (Fiat Auto SpA – copyright reserved)

It is vital that the *high rpm*/advance setting is checked. Set the distributor to give the max stated setting at the relevant speed, and tolerate a certain amount (perhaps +3–5°) excess advance at idle on a distributor with worn springs. Replacement parts for some of the distributors are no longer available (the dreaded 'NLA'!), so if the advance curve is deemed to be totally unsatisfactory, the search would be well and truly on for a better secondhand replacement – or a company able to remodel the unit to specification.

The advance requirement for any engine mirrors the torque characteristic. For example, when torque is low (at low rpm or part-throttle), the advance requirement is high. When torque is good, the advance requirement is lower. Giving an engine 'more advance' (GCT have never used more than 34–37° at full-throttle on any TC) will not necessarily lead to a power increase; indeed, if the figures stated need to be exceeded to give a good torque, the problem with the engine is probably not ignition-related at all, but more likely to be incorrect fuelling, cam timing wrong, low CR, or poor volumetric efficiency.

The reason for the coincidental relationship *(see Case History No 5)* between low advance and good torque is simple: good torque is generated when there is minimal contamination of the cylinder on the intake phase by exhaust residuals. This uncontaminated mixture burns well and does not need to be ignited excessively early. Minimal contamination occurs when good 'blowing down' or scavenging of the cylinder takes place – due to a good match of engine speed, cam/inlet tract characteristics and high volumetric efficiency. Engines with forced induction behave like their normally aspirated cousins until the manifold pressure reaches 1bar absolute (atmospheric pressure – *ie* 0bar-gauge). Then the scavenging process is so effective that high advance is no longer needed (and indeed will lead to engine damage) and as the charge density increases (at higher boost) the advance requirement drops progressively – except that more advance will still be needed at high than at low rpm because of the crank angle/combustion speed relationship mentioned earlier.

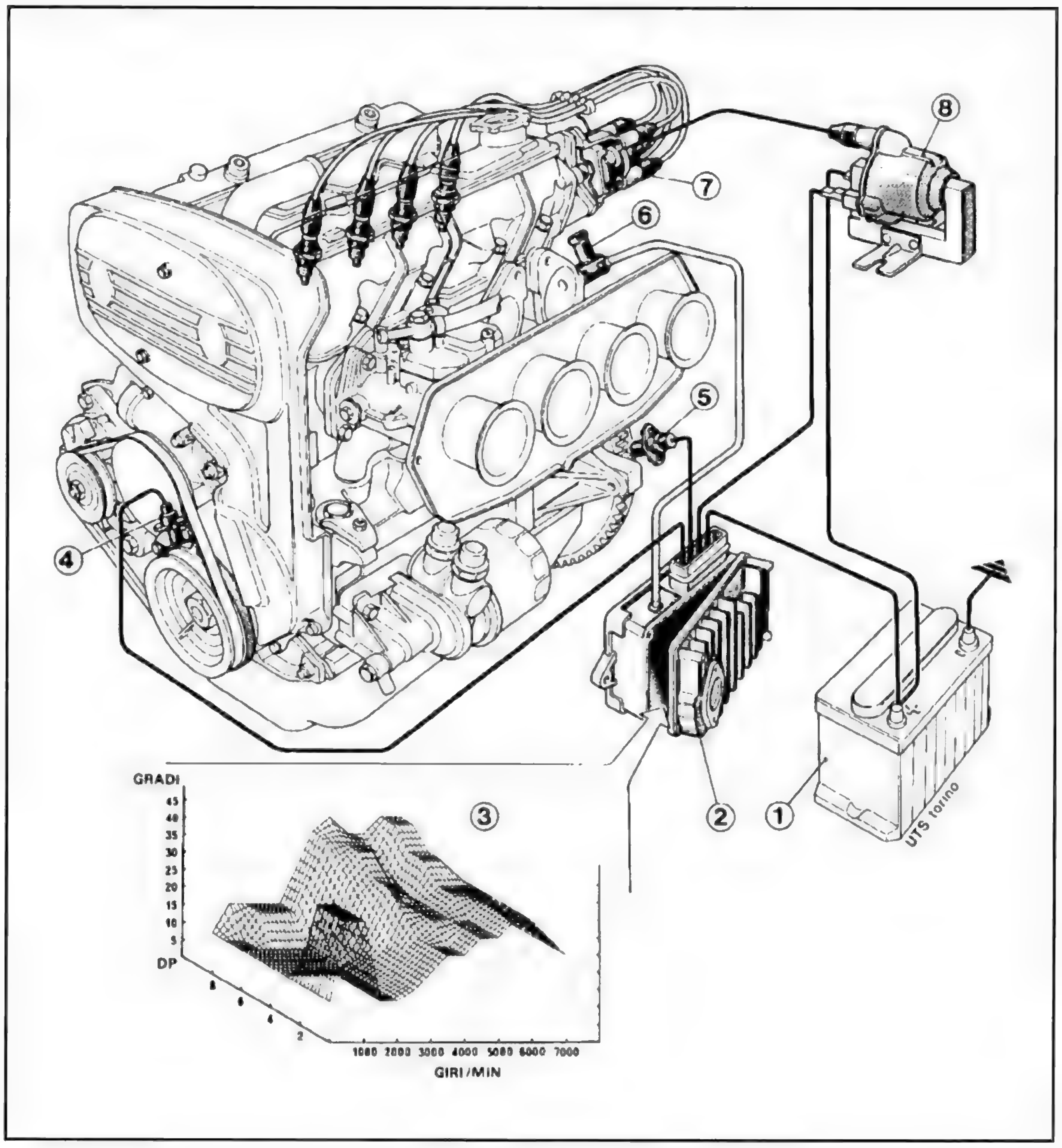
Remodelling a bobweight-type distributor to suit an engine is tedious in the extreme. The engine has to be run on a dynamometer from the lowest speed at which it will hold full-throttle under load to the highest speed/full-throttle position. At various speeds, the distributor has to be swung a few degrees each way to determine the optimum position for best torque. Notes are taken of the ignition timing, and the distributor is then remodelled on a distributor testing machine to reproduce the curve required. To increase the advance, the springs are lightened or bobweights made heavier by welding. To increase the maximum advance, the stops are modified.

GCT are often asked: 'Is the standard centrifugal distributor OK?'. The short answer is that provided it meets the criteria of 34–37° @ 3500rpm (124 CSA) or 5500rpm (others) and produces 10–12° at tickover (possibly as much as 4–5° excess with worn springs) then it should be *left alone*. To properly modify it could require as much as three hours of bench dyno time (on a computerized rig, for repeatability at the right temperatures) and at least the same again on a distributor-testing machine. If the working power band of the engine, in competition, is 5000–7500rpm, for example, this work would be a complete waste of time since the distributor would be producing 34–37° all the time anyway. For an 'all-out' rally engine required to go as low as 4000rpm, some modification of the distributor *might* produce a torque improvement mid-range, but offset against this is the fact that there are probably cheaper ways of optimizing the torque – *eg* bigger valves, cam swap, etc.

Over-advanced ignition leads to pre-ignition and detonation (even if jetting/inlet air temperature are OK) because the combustion is fighting the piston (on the way up), and over-retarded ignition causes late combustion, leading to overheating of the exhaust valve and manifold.

Some distributors incorporate vacuum advance. Vacuum exists in the inlet manifold until full-throttle operation. At part-throttle the ignition is usefully advanced to help 'iron-out' the torque curve. A vacuum diaphragm (very often old ones are ruptured, or too badly worn to work effectively) moves the distributor baseplate to perform a similar function to the centrifugal weights. Vacuum advance can be highly effective.

Ignition timing *must* be checked with a strobe – preferably one with a variable-delay mechanism (Snap-On is one of the best) so that a TDC mark on the block and front pulley is all that is required. At any speed the delay system can be used to align the two marks and the appropriate



11/7: Schematic of the Digiplex electronic ignition system. Note that the drawing of the control unit is wrong – viewed as shown the vacuum tapping should be on the right-hand side of the connector. (Fiat Auto SpA – copyright reserved)

1: battery
2: electronic module
3: mapping of spark advance ignition memorized by module (2)
4: sensor of crankshaft location
5: rpm sensor
6: vacuum connection
7: ignition distributor
8: coil

degrees of advance recorded. If a variable-delay strobe is not available the front pulley must, at very least, have two marks for 10° (at tickover – approx 800rpm) and 34–37° (36° would be suitable enough) for higher rpm. When using the strobe, the vacuum advance must be disconnected.

In the absence of hard data on the vacuum system, check that it advances the ignition a few degrees, *ie* 5 or so, at tickover. (The Volumex vacuum unit adds a maximum of 15°±2.)

It is worth noting at this stage that if CB points are incorrectly set, they will upset the ignition timing; always set the dwell before the timing.

Mapped systems: Digiplex

These were first seen on the 130TC, Delta/Prisma 1600.

A mapped system contains a computer capable of selecting the optimum advance requirement from a mass of stored data, for any given engine speed/load condition. To monitor load, the system requires either a vacuum/rpm sensor *(see Digiplex layout)* or a throttle potentiometer/rpm sensor *(Weber injection – see Case History No 5)*. A conventional coil is used, and the current to the coil is generated by a similar transistorized switching device to that used in the electronic systems mentioned previously.

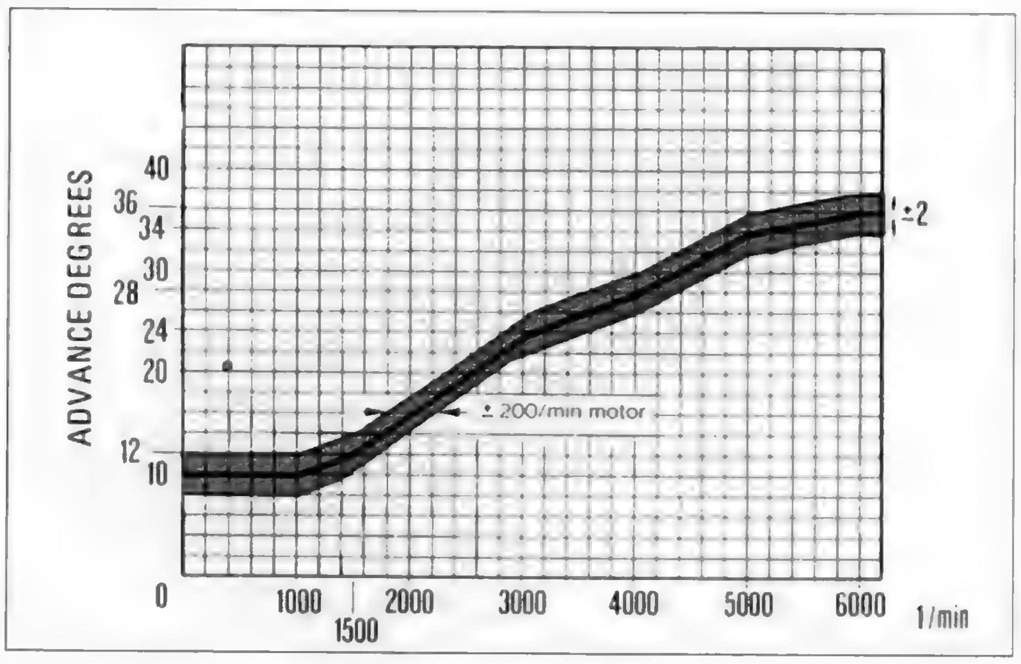
The three-dimensional map (**11/7**) for the 130 TC of engine speed (GIRI/MIN), manifold vacuum (DP) and advance (GRADI) demonstrates how the Digiplex determines which of the 512 stored values is appropriate to the engine's needs. The crank front and flywheel sensors are merely electromagnetic types, and a transducer in the inlet manifold picks up the vacuum condition. The position of the distributor does not control the 'static' timing of the system – this is established by the relative position of the two TDC reference pegs on the front pulley and the front sensor.

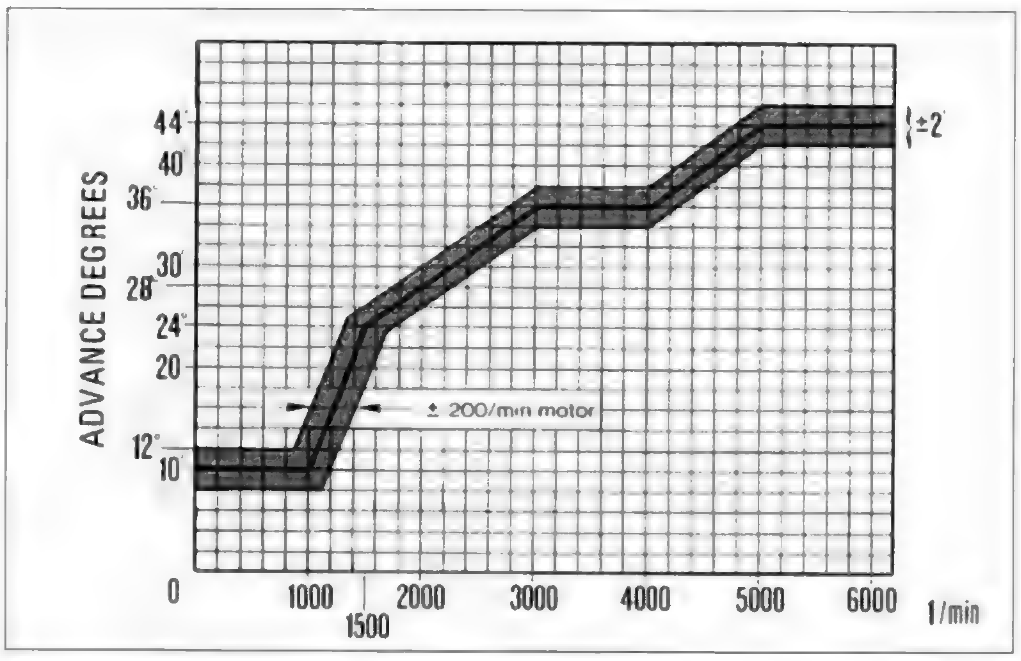
The main advantages of Digiplex over the earlier transistorized systems are:

1 No mechanical wear or moving parts
2 No errors due to coupling of distributor to crankshaft
3 Superior accuracy – hence reduced emissions and smoother torque curve

Digiplex useful data

1	Gearbox-mounted rpm sensor resistance		612–748Ω at 20°C
2	Gap: rpm sensor – flywheel teeth		0.25–1.3mm
3	Gap: front pulley TDC sensor		0.4–1mm (the sensor should be exactly opposite the peg at TDC)
4	Coil data: (ignition off)	primary resistance	0.31–0.38Ω at 20°C (connect across 2xLT terminals)
		secondary resistance	3330–4070Ω at 20°C (connect across HT and one of the LT terminals)
5	Rotor arm resistance		800–1200Ω
6	Advance at idle (800rpm without vacuum)		10°
7	Max advance (without vacuum)		34–36° @ 5500 (no load)

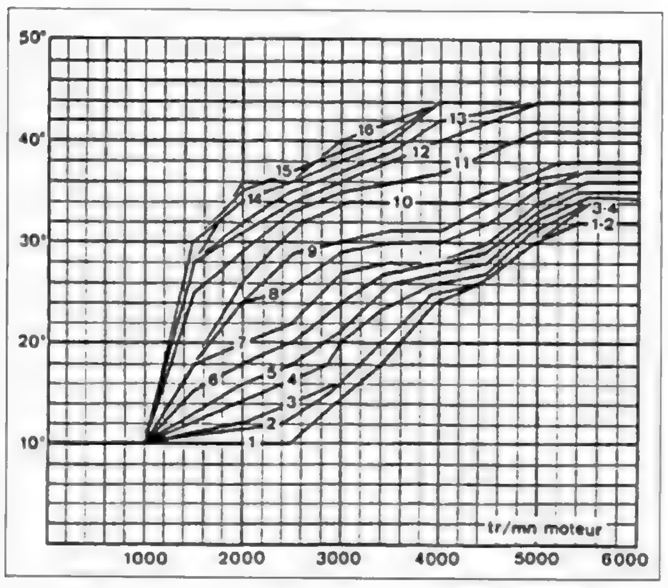




11/8, 11/9: Characteristic advance curves for different vacuum values in inlet manifold: left, 0.026bar/20mm Hg; right, 0.400bar/300mm Hg. (Fiat Auto SpA – copyright reserved)

Mapped systems: Microplex

This is the system used on the Delta 1600HF Turbo (carburetted) and functions in a similar fashion to the 130 TC system except that it is programmed to reduce the advance as the manifold pressure exceeds 1bar (absolute).



11/10: Microplex advance curves versus RPM at various manifold pressures.

Pressure valve (bar)	Advance curve
0.565	1
0.498	2
0.432	3
0.365	4
0.299	5
0.232	6
0.166	7
0.09	8
0.03	9
Vacuum (bar)	
0.03	10
0.09	11
0.166	12
0.232	13
0.299	14
0.365	15
0.432	16

The ignition is designed to cut out at 0.6bar pressure; the turbo is pre-set to a maximum of 0.52bar pressure (7.6lbf/in²).

Microplex useful data

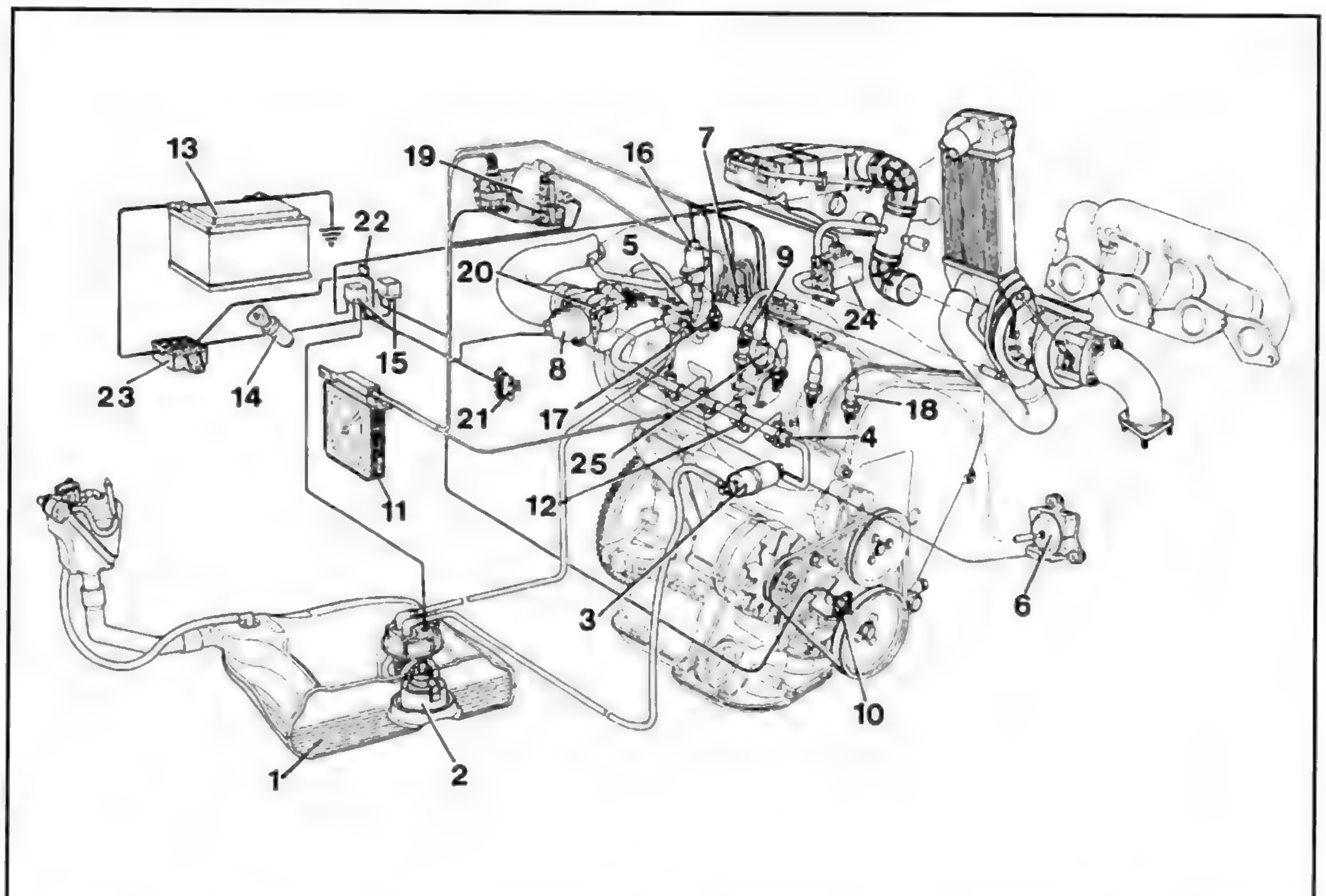
Same as Digiplex except maximum advance (without vacuum, no load) should be 32°±2° @ 5500rpm.

Safety precautions with Digiplex (and other mapped systems):

a Do not start vehicle with battery cables disconnected
b Do not use a 'fast-charger' to start the engine
c If vehicle is baked in a paint-drying oven where the temp exceeds 80°C, remove the control unit
d Do not undo the connector from the control unit when the engine is running
e Always disconnect the earth battery terminal before welding on vehicle

Weber injection/ignition (IAW)

The later generation of TCs, *eg* Delta turbo 1.6 *ie*, Delta-Prisma 4WD, Delta

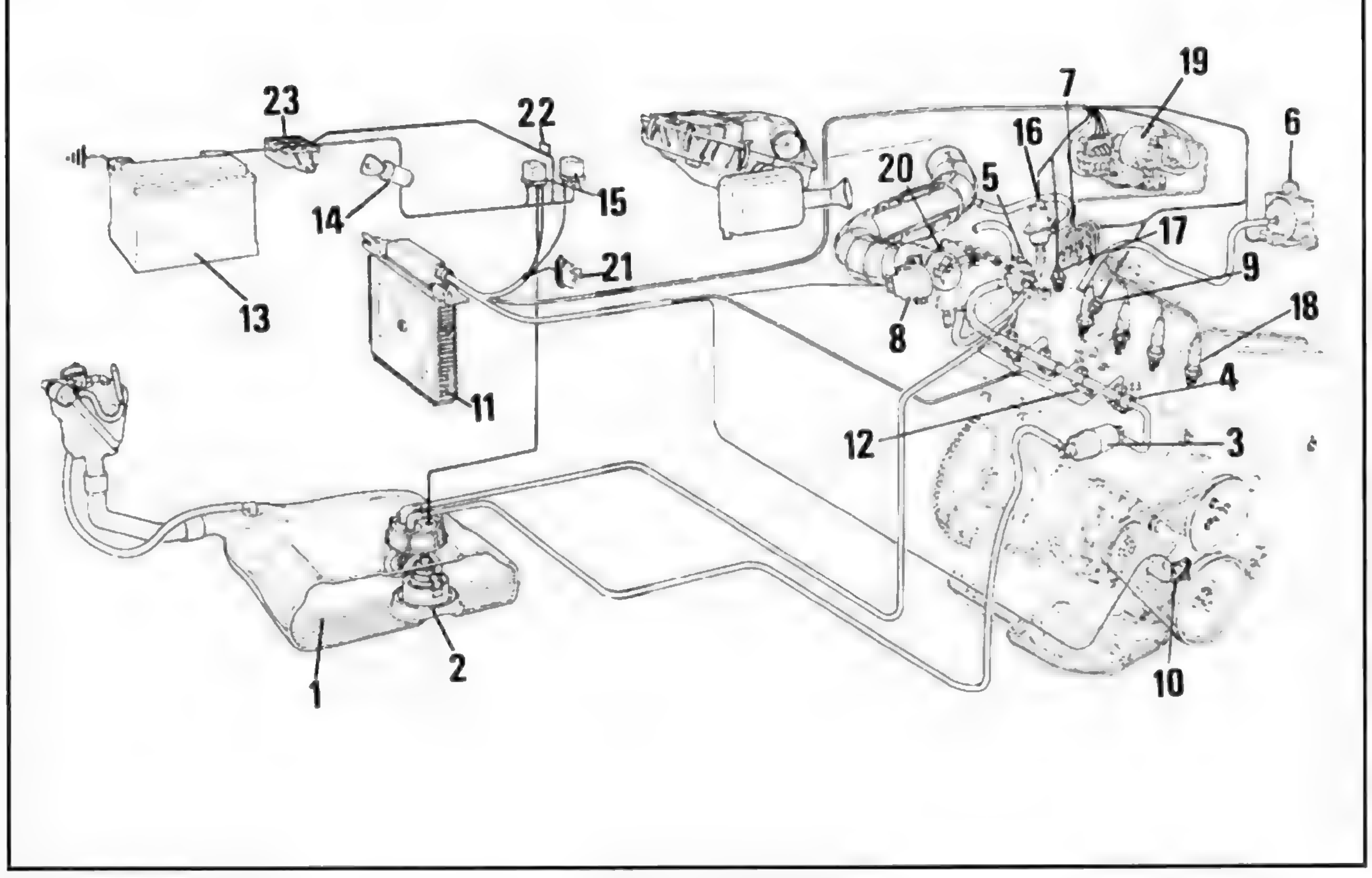


11/11: Diagram showing IAW injection/ignition system – Delta 4WD. (Fiat Auto SpA – copyright reserved)

1: fuel tank
2: electric fuel pump
3: fuel filter
4: fuel inlet
5: fuel pressure regulator
6: absolute air pressure sensor
7: HT distributor with injection timing sensor
8: butterfly valve position sensor
9: inlet air temperature sensor
10: RPM and TDC sensor
11: electronic control unit
12: injector 13: battery
14: ignition switch
15: injection/ignition relays
16: additional air solenoid valve for automatic adjustment of engine idling
17: coolant temperature sensor
18: spark plugs
19: ignition unit
20: butterfly valve
21: diagnostic socket
22: WII system protective fuse
23: vehicle electrical system connector
24: overboost solenoid valve
25: detonation sensor

11/12: Diagram showing IAW injection/ignition system – Prisma 4WD. (Fiat Auto SpA – copyright reserved)

1: fuel tank
2: electric fuel pump
3: fuel filter
4: fuel manifold
5: fuel pressure regulator
6: inlet air absolute pressure sensor
7: HT distributor with injection timing sensor
8: butterfly valve position sensor
9: inlet air temperature sensor
10: RPM and TDC sensor
11: electronic control unit
12: injectors
13: battery
14: ignition switch
15: injection/ignition relays
16: additional air solenoid valve for automatic adjustment of engine idling
17: coolant temperature sensor
18: spark plugs
19: ignition unit
20: butterfly valve
21: diagnostic socket (located near injection control unit connector)
22: IAW system protective fuse
23: vehicle electrical system connector



4WD HF turbo, Tempra 2*l ie*) adopted an integral ignition/injection control system, with the layouts basically split between n/a and turbocharged engines.

The advance characteristic for the normally aspirated fuel-injected versions is not greatly dissimilar to that of the Digiplex system in that the advance up to 40°(±2°) can be 'dialled in' according to engine speed and manifold vacuum. The turbocharged Delta 4WD and similar systems are significantly different in that when the manifold pressure reaches approximately atmospheric (1bar absolute) virtually the whole advance curve is lowered, *eg*:

TOTAL ADVANCE ° AT RPM												
	500	1000	1500	2000	2500	3000	3500	4000	4500	5000	5500	6000
0.92 BAR ABSOLUTE PRESSURE	8	23	22	25	27	29	31	30.5	32	34	34	33.5
1.38 BAR ABSOLUTE PRESSURE (5.5LBF/IN² BOOST)	8	10	14	13	14	15	22.5	22.5	24	25	26	26
1.80 BAR ABSOLUTE PRESSURE (11.8LBF/IN² BOOST)	5	5	5	4	4	5.5	8	12	14.5	16.5	18	10

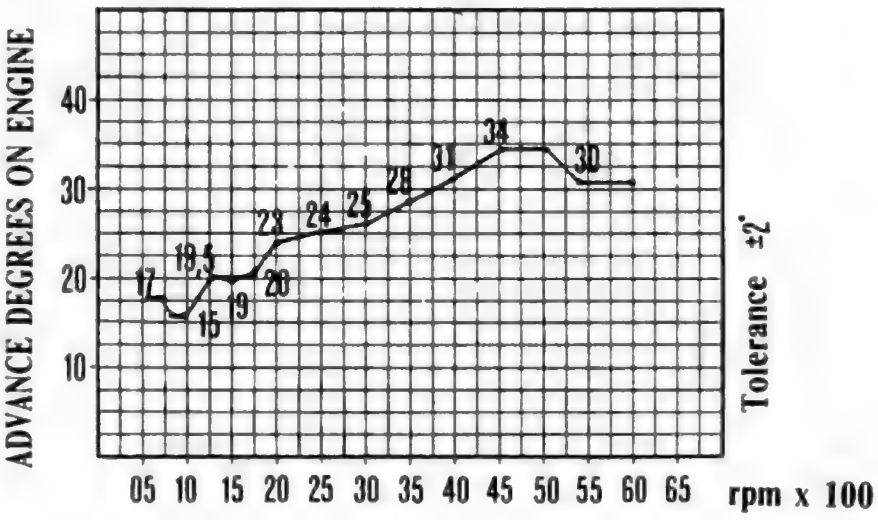
Exhaustive practical tests by Fiat/Lancia were required to determine the optimum values for ignition advance for these various systems to ensure the best combination of economy, power and minimum emissions. These values are stored in the ECU (item 11); speed and load are fed into the ECU computer and the advance at any condition is subsequently determined. Speed and TDC position are monitored by means of a single sensor on the crank front pulley (which, unlike the Digiplex system, has four teeth compared with two). The sensor is a reluctor, which generates a single-phase AC current as the teeth on the pulley pass by; the frequency of the current indicates the engine speed. Load is measured by means of an absolute pressure sensor (0bar-gauge = 1bar absolute atmospheric pressure), which produces a variable, continuous voltage proportional to the manifold pressure. Unlike the Digiplex, the distributor contains a reluctor, which acts as purely a timing sensor for the injection phase, so that the ECU can identify the operating stage for each cylinder: the rotor and cap are the only parts of the distributor connected with the ignition system!

The amplifier module for the ignition system is located on the heat sink of the coil unit.

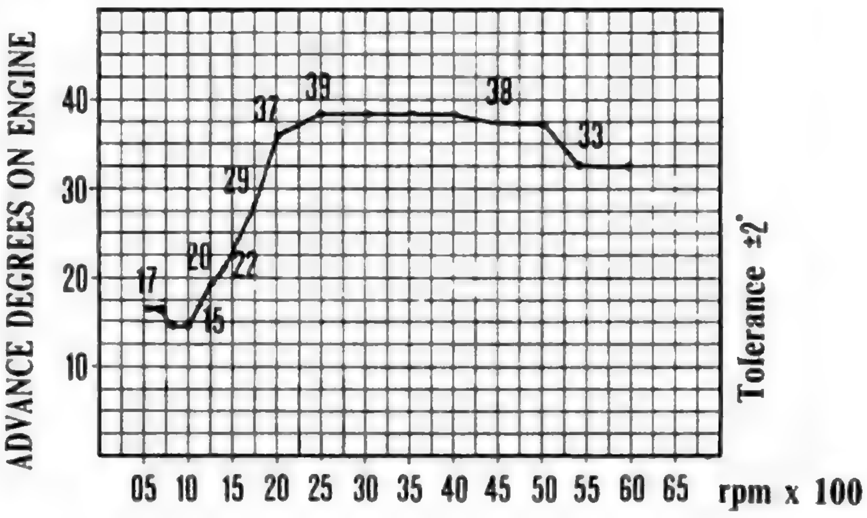
Useful data Delta/Prisma 4WD (n/a or turbo)

Idle speed ignition advance	2*l* turbo (8v) models 18°±2° 2*l* Prisma 4WD 15°±2°	
(If the ignition setting is wrong the rpm/TDC sensor is incorrectly positioned or there is a fault with the ECU)		
TDC/RPM sensor resistance	618Ω–748Ω @ 20°C (same as Digiplex front sensor)	
Sensor gap	0.6–1.2mm	
Distributor timing sensor resistance	758Ω–872Ω @ 20°C	
Resistance gap	0.3–0.4mm	
Coil resistance	primary	0.415Ω–0.495Ω @ 20°C
	secondary	4320Ω–5280Ω
Rotor arm resistance	1000Ω	

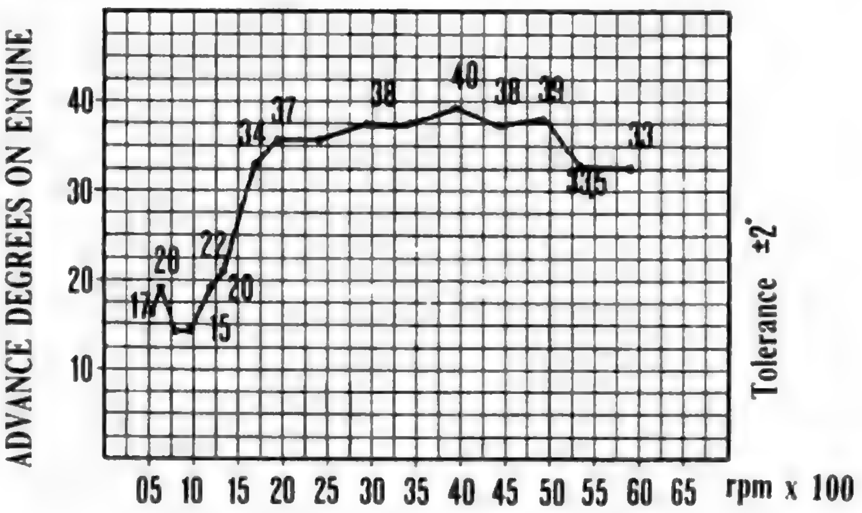
at an absolute pressure of 0,18 bar (135 mmHg)



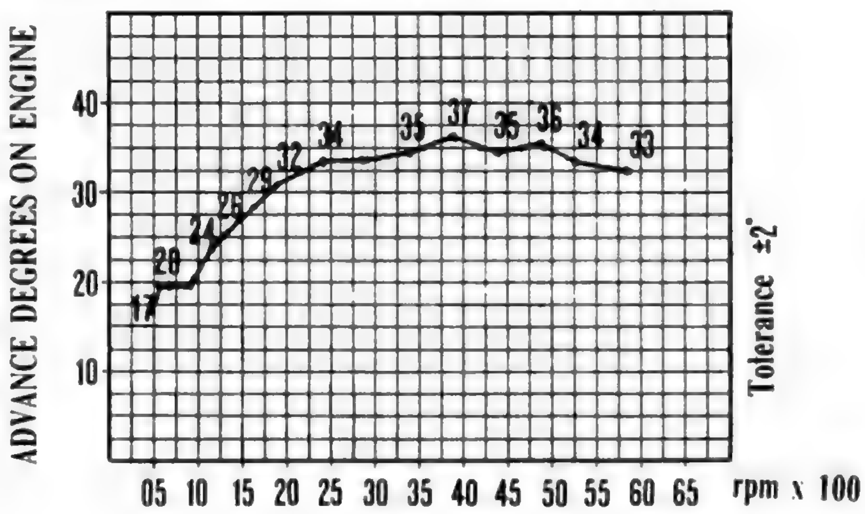
at an absolute pressure of 0,299 bar (225 mmHg)



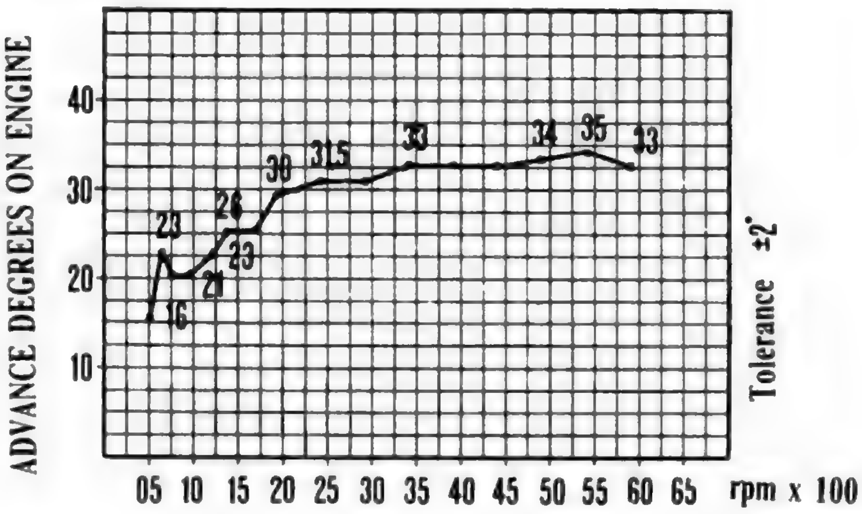
at an absolute pressure of 0,43 bar (321 mmHg)



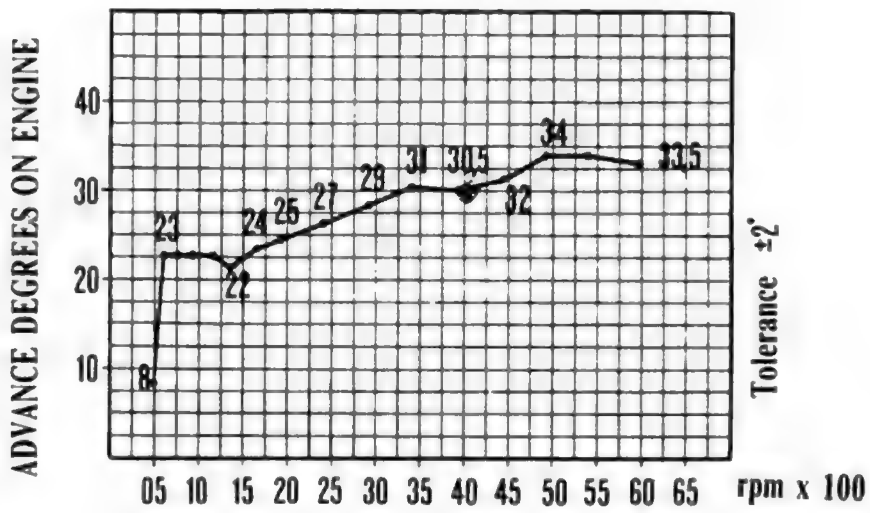
at an absolute pressure of 0,54 bar (405 mmHg)



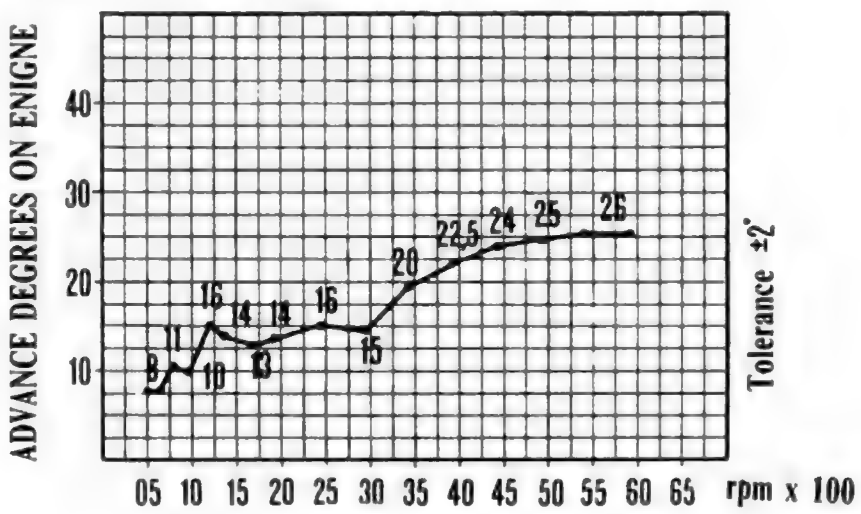
at an absolute pressure of 0,70 bar (525 mmHg)



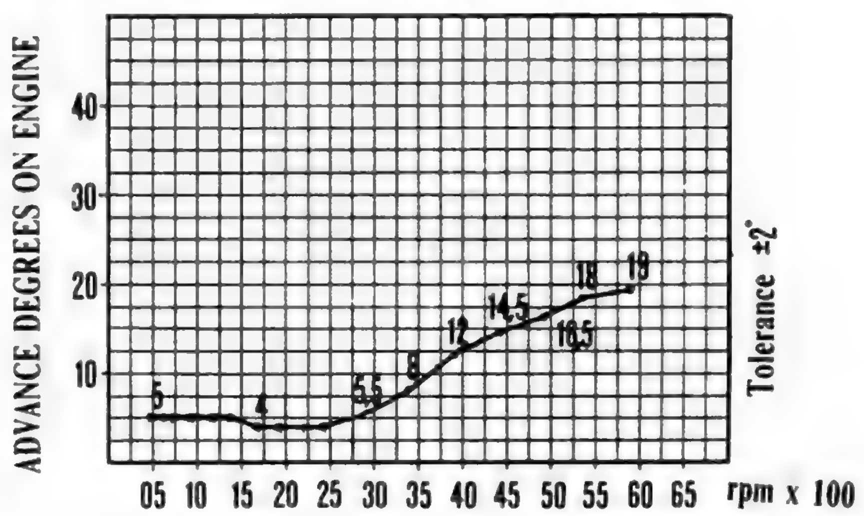
at an absolute pressure of 0,92 bar (690 mmHg)



at an absolute pressure of 1,38 bar (1035 mmHg)



at an absolute pressure of 1,80 bar (1350 mmHg)



11/13: Ignition advance against vacuum/rpm for Delta 4WD (turbo). (Fiat Auto SpA – copyright reserved)

Detonation sensor

Fuel-injected, turbocharged models are fitted with a 'knock' (detonation) sensor. This comprises a sensor bolted onto the cylinder head and connected to the ignition/injection control unit in order to adjust the intensity of the vibrations (knocking) caused by the detonation in the combustion chamber whilst the engine is running.

If this is the case, the sensor informs the ignition/injection control unit so that it can quickly reduce the engine ignition advance values. The reduction of the advance values takes place when the system recognizes 'engine knock' due to detonation as distinct from normal combustion.

The advance curve for a given engine load is reduced by around 5°. If the detonation should still persist, the advance is further reduced by 5° at a time up to a maximum of 15°. After a certain number of operating cycles without knocking, the advance is then gradually reinstated to its original value. The advance curve cannot be reduced by more than 15° in relation to the original curve according to the engine load conditions, supercharging pressure and engine speed.

This device is essential in safeguarding the life of the engine as detonation can very easily occur whilst the engine is being pressure-charged.

Testing/fault-finding

Because the fuel system is connected to the ignition system on these models, it can be difficult to establish which is faulty when a problem occurs. Checking the ignition system starts with these simple steps:

1 Open bonnet and switch on ignition. If a spark is clearly heard inside the distributor cap it means the ignition module and ignition control system within the ECU are probably OK. It is cheaper to replace the amplifier module (on the coil unit) than the ECU if there is no spark, but first test the coil, as a faulty coil can damage the amp module.

2 If checking the sensor resistances, disconnect their respective terminals first.

3 The operation of the coolant and intake temperature sensors, butterfly valve potentiometer (position sensor) and relays can all upset the ignition system. Realistically, space precludes a full (and practical!) summary of the test procedures involved (which are mostly continuity checks); the advice of a competent Fiat/Lancia dealer should be sought.

General ignition checks – all systems

Distributor cap:	Check for cracks, chronic oxidation of segments
Rotor arm:	Check (carefully) for cracks and correct fitment (as cap)
HT leads:	Check that resistance is within limits, plug caps secure
Coil:	Ensure the coil tower is clean

A 'first-line' fault-finding check on ignition systems is to crank the engine with a plug lead disconnected from the plug and held (with heavily insulated pliers) approximately 4mm from an engine earth and examine the intensity of the spark. Any system in good working order would produce a thick (2mm) royal-blue spark which is capable of arcing across the 4mm gap. Extend the gap to as much as 10mm and the spark should easily jump the gap (it will become thinner). Make sure that the plug lead is held well away from any fuel source, and for personal safety, wear rubber-soled shoes and do not touch any part of the body of the vehicle (the human body acts as a very good conductor at 15KV-plus!).

Never crank an engine with electronic ignition if the main HT lead from the coil to the distributor is not properly connected – the amplifier unit may blow.

Summary of distributor layouts

Type	Position
130TC, late (Digiplex) 105TC	inlet cam end drive (peg)
Lancia Monte Carlo	inlet cam top drive (gear)
124 Sport 1608/124 CSA	ex cam top drive (gear)
Argenta 2*l ie*	ex cam top drive (gear)
Delta/Prisma 1600, 1600 HF turbo (carb)	inlet end drive
Delta HF turbo 1600 *ie*, 1600 *ie*	exhaust end drive (at front of car due to reversed port layout)
2*l* 8v turbo (except Thema)	exhaust end drive (at front of car due to reversed port layout)
Thema turbo (2*l ie*)	inlet end drive

Author's note: I have included this short list because of the number of 'head swaps' undertaken by clients!

Spark plugs

(*Author's note:* I have confined myself to a description of NGK plugs suitable for TCs since these are the types now exclusively used by GCT.]

Explanation of symbols (using the BPR6ES plug as an example):

B	P	R	6	E	S
Thread dia B=14mm	P=projected nose insulator	Resistor	Heat range	Thread reach E=19mm	S=copper core

Notes:

B All TCs have the same plug thread diameter

E All TCs have the same plug thread reach

R These type of plugs have a resistor inside the plug core to reduce interference with communications equipment and engine management systems

6 Heat range number. The types available vary from codes 2 (hot) to 12 (cold)

S Other useful variations in this code include:
Y – V-groove plug (requires lower voltage and reduces emissions)
V – Gold palladium centre electrode – gives long service life in race use (1mm dia)

GCT recommendations:

Fast Road	BPR6ES (these plugs are available with V-groove electrode as standard)
St II, St III	B9EGV
Full Race (n/a)	B95EGV (for standard engines use standard plug types)
Turbo GpA +	B10EGV (cold)
Recommended gap	
	0.8mm
Recommended torque	
New gasket	½–¾ turn (3.4–4kg m)
Old gasket	1/12–⅛ turn

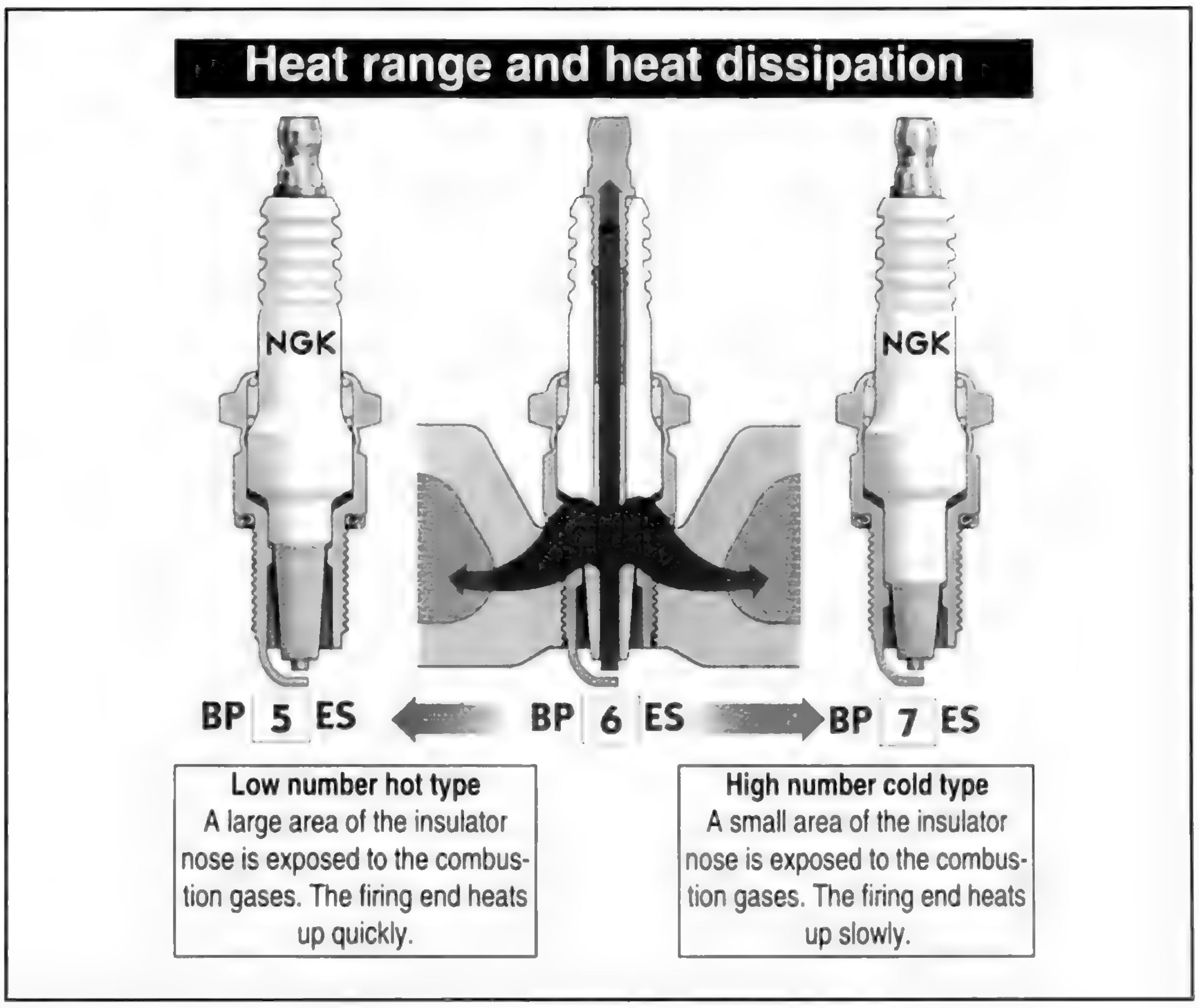
Firing end appearance (on plugs after they have been run at full-load/full-throttle for at least 15sec):

Optimum temperature area (450°–870°): The insulator in good condition will be light grey-brown. Fuel-injected engines (not fitted with catalytic converter) will tend to be at the leaner end of the colour band (grey), carburetted engines may tend towards brown. The outer ring of the plug is grey (injection) to black (carbs). Fuel-injected engines will consistently run at 4½% CO under full load whereas carburetted types may need 4½–5½% CO (and high-boost turbos).

Overheating area (870°C-plus – pre-ignition danger):

The insulator is white and sometimes blistered (also the outer ring).

Fouling area (less than 450°C): Carbon accumulation on the insulator nose forming a leakage path to earth (common with unleaded fuel). Prolonged driving at idle on low speed can lead to problems (plug idle temp is around 150°C–250°C).



11/14: Showing the effect of spark plug insulator nose exposure on speed of heat build-up.

CHAPTER 12

FORCED INDUCTION

Turbocharging *versus* supercharging

Comparison

Most readers will be well-acquainted with the two methods used by Fiat/Lancia to achieve forced induction – supercharging and turbocharging. The debate as to which is 'better' has raged back and forth for years, and it is worth considering some of the less well-known aspects of the two systems. The supercharger (in this case the Volumex unit) is driven off the crankshaft; the turbo is driven by exhaust gas, so in terms of their effect on thermal efficiency (which, in essence, is merely a measure of how much power can be derived from a given amount of fuel) the turbo produces a better result since it utilizes energy from the waste gas to drive the compressor. Additionally, the centrifugal compressor used in turbochargers is a more efficient air pump than the Roots-type blower. Because of this, the turbo unit will produce far greater airflow and pressure than an equivalent supercharger, and it has an inherently better mechanical efficiency because the moving parts are lighter – reducing inertia and friction.

From the discussions of volumetric efficiency earlier, it will be clear that the purpose of forced induction is to improve the filling of the cylinder to raise torque output. The effect of forced induction is to raise the pressure of fuel-air mixture in the inlet manifold so as to raise the charge density, *ie* squeeze more of it into the cylinder. Because of the limited pressure differential across the valve throat, between the cylinder and atmosphere, it is obvious that any increase in this differential will lead naturally to an increase in torque, but the curious nature of the airflow demand of the reciprocating engine highlights an interesting anomaly that exists in the use of the turbocharger on such an application.

It may seem paradoxical to say it after all the success that has been achieved with such systems, but the characteristics of the centrifugal compressor used in turbo systems are poorly matched to the requirements of a petrol engine. Compare the airflow demand of a jet engine with that of a piston-type: in a jet engine combustion chamber there are no valves between the intake and exhaust. The compressor (early British jets used radial, or centrifugal compressors, rather than the modern axial design) is connected to the turbine in the same way as a turbocharger, and blasts air continuously into the combustion chamber – which is burning at all times. The result is a combustion process known in thermodynamics as 'steady flow'. The compressor and turbine characteristics determine the output and rate of the combustion process, and moreover the unit is required to operate within a narrow speed band – take-off/landing and cruise. These are ideal conditions for a power unit with massive inertia (a function of the mass, diameter and rotational speed) – one which it is impossible to start/stop or accelerate quickly.

Now look at the turbocharger attached to the reciprocating engine. Essentially, the unit is rather like a jet engine with a 'remote' combustion chamber – where the compressor output is fed through a series of reciprocating cylinders, each of which fires only once every two crankshaft revolutions, and the exhaust is then forced to circumnavigate the valve and exhaust manifold before it can reach the turbine and perform useful work. The steady-flow unit is now required to work under 'non-steady-flow' conditions. Added to this is the obvious fact that in a petrol engine the crankshaft speed varies enormously in range and is constantly changing, as indeed is the load on the engine, and most importantly the airflow demand. Certainly, when the demand for torque is high, the turbocharger is admirably suited to the task. The problem arises when the demand is for limited duration. The turbo components have high inertia due to their very high rotational speed (100,000rpm-plus on full power). Having been spun up to very high speed, the inertia effect makes the unit very reluctant to slow down when the torque demand drops. Conversely, when the torque demand suddenly increases, the inertia fights against the process and the unit is hard to 'get going' again. Add to this the fact that to produce high boost on an engine with a high airflow demand (big valves, long-duration, high-lift cams) requires a bigger turbo – or the demand from the engine will outstrip the flow characteristic of the turbo (for example, the turbo on the excellent Lancia Delta 1.6 HF turbo *ie* cannot deliver more than 22lbf/in^2 boost on full-throttle if the engine is even mildly tuned). The turbo, because of this inherent mismatch, is constantly underpressurizing/overpressurizing the engine; the root cause is clear to see: unlike the jet engine with its integral combustion chamber, the turbocharger is not directly 'reactive' to the speed of the combustion process.

None of this is new, of course, and manufacturers (and tuners) have devoted enormous resources to trying to overcome this inherent mismatch of capabilities, but it has never, and never will be completely cured. Diesel engines, with a crank speed range of far less than the petrol unit and a slower combustion process, are eminently more suited to turbo conversion (especially commercial units fitted with a multi-speed gearbox to keep the engine operating within a narrow speed band). However, complex systems of compensating devices ensure that, certainly in the lower-middle to upper speed range of the engine, turbochargers (these days!) work surprisingly well.

Like the turbo, the supercharger produces an airflow/pressure characteristic which is speed-dependent, the crucial difference being that the 'blower' is connected directly to the crankshaft and the system is thus speed-sensitive. The very nature of the Roots blower, with its

interlocking rotors and the leakage path between them, makes it far less efficient than the turbo. It is not practical to spin a unit at very high speed because of the inertia load this would impose on the drive system and its effect on the accelerative rate of the engine, and because of the 'surge' characteristic generated by this type of blower at quite modest boost: when the pressure generated inside the casing is excessive, back-leakage between the rotors can lead to shock-wave damage to the mechanism. To produce high pressure requires a bigger supercharger, which is clearly undesirable for production car applications.

Engine effects

Both systems raise the charge density in the cylinder – this raises the imep and increased torque follows naturally. It follows that, since the cylinder (and clearance volume) is packed with fresh charge, better scavenging of the cylinder, *ie* removal of exhaust residual, takes place than with the n/a engine – so not only is there more charge, it is less contaminated.

The high peak pressure leads to a greater load on the engine *(see Chapter 2 – Stress)*. In particular, the thermal load is significant. The problem of overheating the engine (particularly the exhaust valves) and detonation damage are issues which must be carefully considered at the design stage (or if a conversion is considered). Both types of forced induction lead to an enhancement of the torque curve against an equivalent n/a engine in that there is 'more of it for more of the time'. Reinforced pistons, stronger gaskets, sodium-cooled valves, bronze guides, special exhaust valve seats and superior lubrication systems are all factors to be considered. The turbo system exhaust manifold should be made especially strong because of the phenomenal heat generated between the two components, and systems employing a 'blow-through' carburettor (Delta HF turbo 1600) and fuel injection need to be designed to cope with the high manifold pressure.

Head modifications, exhaust and camshafts for forced induction

Forced-induction engines respond in the same way as normally aspirated to porting, valve and seat modifications. The influence of interference effects in the inlet tract is lessened because of the over-pressure present, and reverse flow of exhaust gas is reduced – although a turbo 'off boost' (at low throttle) suffers from back-pressure between the turbo and exhausting cylinder. Long exhaust branches on a turbo system are undesirable since their objective is to force the high-energy exhaust gas (and pressure pulse) through the turbine as soon as possible. With a supercharged system the obvious choice is similarly a 4–1, but with long primary pipes – to enhance the scavenging effect in each exhausting cylinder (an 'interference' system is not needed) – the requirement for high vacuum around TDC at the point of inlet valve opening is far less since the fresh charge is being forced in under pressure.

In the main, the camshafts will dictate the performance characteristics of the pressure-charged engine in the same way as for the n/a type in that the lift and duration will still determine the amount of fuel/air mixture entering the cylinder, and at what point in the load/speed range; but there are a number of unique features relevant to camshaft selection with these types of engine which are worth examining.

Overlap

The extent of overlap, in terms of the LATDC and the period for which both inlet and exhaust valves are held open at the end of the exhaust stroke, is responsible for the effectiveness of cylinder scavenging – contamination by residuals has a disproportionate effect on power loss. Heavy overlapping on a normally aspirated engine leads to poor torque at low rpm, when the exhaust energy is low as it tends to flow back up the inlet tract. If the inlet charge pressure is high (*ie* well above atmospheric) this reverse-flow tendency is inhibited, and such is the case with forced induction, particularly supercharging (due to the better boost-crank speed characteristic), so more torque will result. However, especially at low speeds, there is a risk of excessive amounts of charge being blown straight out of the exhaust and creating the opposite effect. The 'blowing-down' effect gets progressively better at high engine speeds because the time period for which the exhaust valve is held open is reduced. Blowing-down also helps to cool the cylinder, and especially the exhaust valve, reducing the risk of detonation.

Valve timing

Because of the back-pressure problem with turbos, the point of opening of the inlet valve on production TCs tends to be very close to TDC, *eg* cam timing:

HF 4WD in 8/42, ex 42/1
16v Integrale in 8/35, ex 35/0
(timing figures quoted with 0.8mm clearance)

There is no need to open the inlet valve early because the charge enters at higher velocity than on the n/a engine, and it can be closed earlier, too – with the Volumex, inlet valve opening is late to reduce fuel loss out of the exhaust port.

High cylinder pressure means that when the exhaust valve is opened the waste gas exits at higher pressure – so it can be opened later than on an equivalent n/a engine – extending the power stroke and extracting the maximum possible work from it. Because of the effectiveness of the exhaust process (and the fact that the pressurized charge will scavenge the cylinder) the exhaust valve can be closed similarly close to TDC, and the maximum lift can be reduced.

This is not to say that these rules cannot be significantly altered, but if the manufacturer can meet his design targets for torque/bhp at the required engine speeds, there is no point at all in fitting cams any wilder than necessary! Raising the output of a turbocharged engine by altered cam configurations is an interesting subject.
The following is a short summary:

Extended inlet cam duration, high lift:

Worse low-rpm torque (for given boost), better cylinder filling at higher speed-under-load and blowing-down benefits. Adverse fuel consumption.

Extended exhaust cam timing:

Higher turbine speed (due to the stronger pulse present when valve opens early in the power stroke) and thus a stronger boost characteristic at mid-range speeds and beyond (worse below!), added to this the benefits of 'overlapping', as discussed earlier. (The exhaust valve peak lift can be retained at 85–95% of inlet without penalty.) These rules apply also to the Volumex except that early exhaust opening should be avoided.

Compression ratio

By lowering the static CR, more boost can be tolerated without increased risk of detonation – and a higher mean effective pressure will result. High CR – high boost must be avoided. The result of this, as far as turbos are concerned, is that the 'off-boost' performance is poor – equivalent to a very low-compression n/a motor! The supercharger produces boost right from tickover and therefore gives superior performance 'off the line'. As a guide, the CR figures given in *Chapter 2* should be used.

Boost controls

The conventional integral wastegate system is used on all the TCs. With this

system a bleed pipe from the inlet manifold (at boost pressure) forces back the spring diaphragm unit inside the actuator, opening the wastegate valve and allowing some of the exhaust gas to bypass the turbine. The Volumex blower is only rated at 6lbf/in^2 max pressure at 5500rpm, so a simple blow-off valve is incorporated into the inlet manifold (set at 15lbf/in^2) to prevent backfire damage.

Modification of a turbo to raise the boost involves use of a different-rate spring in the actuator – a stiffer item will raise the blow-off pressure. Two-stage (or more) boost simply requires the use of a bleed valve which can be switched to divert pressure from the manifold tapping back into the intake system, thus effectively 'fooling' the wastegate actuator.

Regrettably, GCT do no possess the necessary flow/pressure data on the Volumex to be able to specify the maximum rpm permissible for the blower before surge will occur, but experiments with a 7% increase in gearing (up to 7000 engine rpm) seemed satisfactory – this certainly increases the low-mid-range acceleration.

Certain TC turbo models, *eg* Delta HF 1.6 *ie*, Integrale, incorporate 'overboost'. This is a form of two-stage boost which operates when the throttle is fully depressed and the engine speed is between 2000–5600rpm; it will only operate for a maximum of 30 seconds. A solenoid valve bleeds air away from the wastegate actuator circuit, reducing the pressure in the actuator to atmospheric. The throttle position is monitored by a microswitch and the engine speed/boost by the ECU.

12/1: Volumex blower stripped.
Alloy casing houses heavy-duty front and medium-duty rear ball bearing assemblies, helical gears, steel rotors and seals. Gears are only taper-fit on shafts and correct setting-up (phasing) is vital to ensure correct running clearance. Note that rear seals (only one is shown) are fitted so as to prevent suction of air into blower ie are fitted opposite way round to crankcase oil seals. Front and rear bearing housings contain vent drillings to allow electric pump to evacuate fuel vapour from casing when engine is hot and switched-off. This pump is actuated from ignition circuit and temperature sensor in (standard) auto choke coolant hose. Although standard pumps used can only be sourced through Lancia (and are prone to failure on older models) a Facet solid-state pump can be wired to perform same function. Hard-to-get gaskets can be replaced with silicon gasket. Only rear bearings and F/R seals are available as replacement items (and can be obtained through bearing stockists). Nothing can be done about excessive backlash on gears. Golden rule is: 'if it works – leave alone'!

Charge cooling

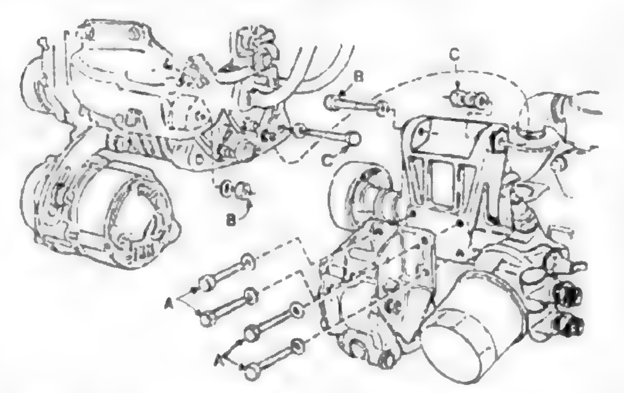
Because the compression of air raises its temperature, forced-induction engines are more susceptible to detonation *per se*. Turbocharged engines suffer from the disadvantage that colossal heat transfer takes place between the air charge and compressor unit, and the unit operates at high boost levels. An intercooler is vital to reduce the charge temperature. The Volumex has no such refinements, but both systems require good ducting of cool intake air to the compressor to keep it at a manageable level. Spray water cooling to the intercooler is highly advisable for high-boost turbos – especially if they are required to run in high ambient temperatures (20°C-plus).

Turbocharger modifications

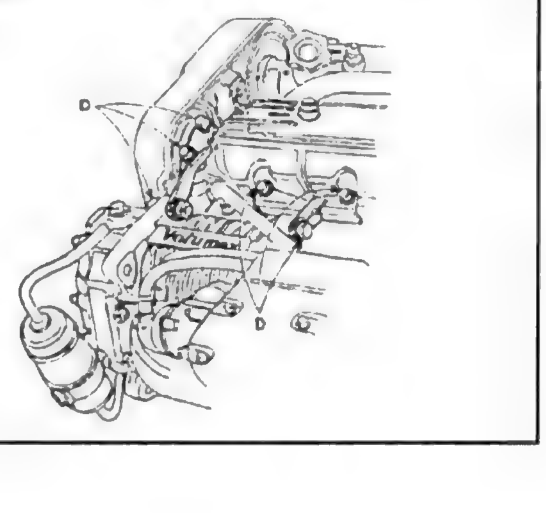
A discourse on the full range of modifications and changes available to the builder of a performance turbocharged engine is a subject in its own right and is beyond the scope of this book. The author has indicated a preference for recommended suppliers at the back of the book for further information on reconditioning, uprating, air circuits, hybrid layouts, dump valves, etc. Turbocharger modification is a job best left to the dedicated expert. The inquiring reader will also find Alan Allard's seminal work on the subject – *Turbocharging & Supercharging* (ISBN 0-85059-494-4) – most useful.

Volumex – useful information and tuning

The Volumex system employs a suck-through carburettor system. The Beta used a Weber 36 DCA 5/250 synchronous-throttle twin-choke carburettor (Pininfarina Spider Europa a 34 ADE 150/4E). Marelli inductive discharge ignition with vacuum advance is employed; ignition data is as follows:
Static advance 10°
Centrifugal advance 25°±2° @ 3100
Max vacuum advance 15°±2°
Resistance winding resistance 730 Ω ±7%
Coil primary resistance 0.44 Ω ±10%



12/2: Supercharging – sketch of servicing instructions.
When refitting blower to engine, strictly follow this tightening sequence:
1 Tighten blower nuts to manifold
2 Slightly tighten items A–B–C
3 Lock B 4 Lock A 5 Lock C
6 Slightly tighten items D first, then lock them



Coil secondary resistance 8400 Ω ±10%
Recommended plugs – Champion N7Y/Bosch W6D/Marelli CW 78LP
Gap – 0.6–0.7mm
Float level (with top cover vertical and gasket in place) – 41mm between gasket and float (measured at the furthest point)

The production Beta Volumex carburettor employs 26mm chokes, which, coupled with the *small carburettor*, severely restrict the torque/power output. If any modifications (porting, race valves, use of alternative cams *eg* Beta *ie* inlet type) are undertaken, it is worth considering use of an alternative carburettor. Raising the airflow through the engine (with the standard carb) tends to overly enrich the mixture and can lead to excessive vacuum advance (and pre-ignition problems), in the same way as a cam swap on a normally aspirated engine invariably requires larger chokes. The supercharged TC, however, must be treated with caution when it comes to setting up an alternative carb. Overly large chokes will cause the supercharged engine to stall badly when the throttle is snapped open because the fuel condenses on the large supercharger 'wetted area' and is prevented from reaching the cylinders. Keeping the choke small retains a very high airspeed, and consequently high pressure drop in the carburettor, thus giving good fuel flow in this transient condition (opening the throttle slowly does not lead to the same dramatic stall effect). Compared with an equivalent n/a engine, the choke size required for a supercharged unit can give up to 80% more power if the carb is big enough – so it is clearly no impediment. GCT have experimented extensively with the DCNF 40 (26–27mm chokes) on the standard manifold and a 45 DCOE (32/33 choke) on a modified manifold and achieved substantial results.

The long inlet tract between the carb and blower on the standard installation creates problems of its own: the fuel drops out of circulation. The standard engine is equipped with a heated manifold to help cure the problem. An electrovalve is also incorporated into the fuel-feed circuit, which, during normal running, allows the fuel to feed the carburettor through a 3mm duct. A bimetallic thermoswitch located in the fuel circuit switches the electrovalve to a 1.4mm diameter fuel-feed orifice when the fuel temperature exceeds 70°C, to decrease the flow of fuel for hot-starting.

When a conversion carb is fitted (such as DCNF, DCOE) the needle valve should be increased in size (210 is a good starting point). Race-type emulsion tubes should be used with jet sizes roughly comparable to those used on the normally aspirated engine *(see Chapter 10) eg* 150 main jet/180 air corrector on a single DCOE. Because of the very strong signal in the secondary choke created by the suction of the blower, huge jets are not needed. The pump jet size should only be increased if good snap-response results are not achieved by varying the choke size and then only increased in 0.05mm steps – if the pump jets are excessively large the engine will over-fuel in the transient mode and fuel consumption will be high!

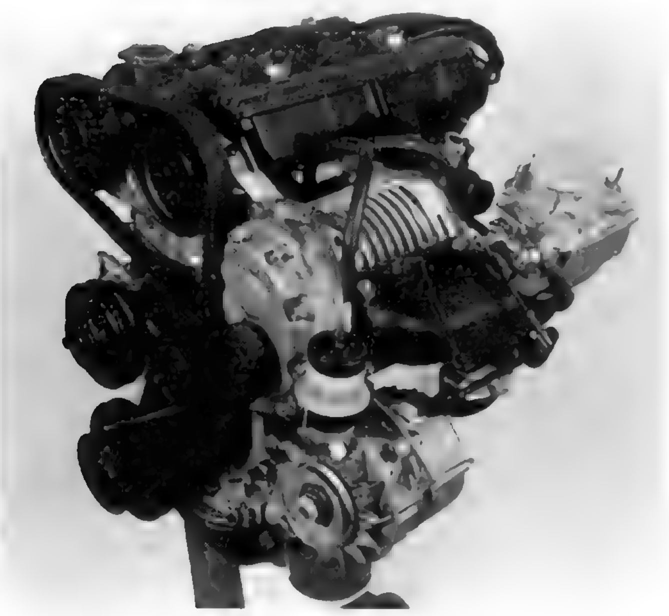
Because the standard and alternative carbs mentioned are all synchronous (*ie* both throttles open together), when setting the idle mixture screws, they must both be screwed out an equal amount.

The vacuum advance can usefully be retained on the Volumex on tuned versions, but check for pre-ignition problems with it connected.

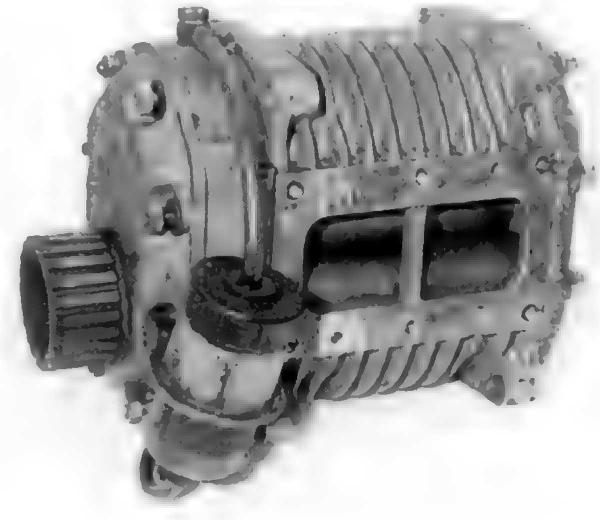
SUITABLE CAM COMBINATIONS FOR VOLUMEX, TURBOS

ENGINE	USE	IN CAM	EX CAM	NOTES
VOLUMEX	FAST ROAD	BETA *ie*	Std	40 DCNF/DCOE 27–29mm choke
	RACE	GC 3A	BETA	45 DCOE 30–32mm choke
DELTA HF 1600	FAST ROAD	130 TC	Std	Std boost
	RALLY (CLUB)	130 TC	130 TC	15–18lbf/in² boost
	Gp A	GC 3A	130 TC	18–20lbf/in² boost
2/8v TURBO	FAST ROAD	as DELTA HF 1600		
	Gp A (ultimate)	Hybrid (see notes)	GC IIIA or 1608 124 type	

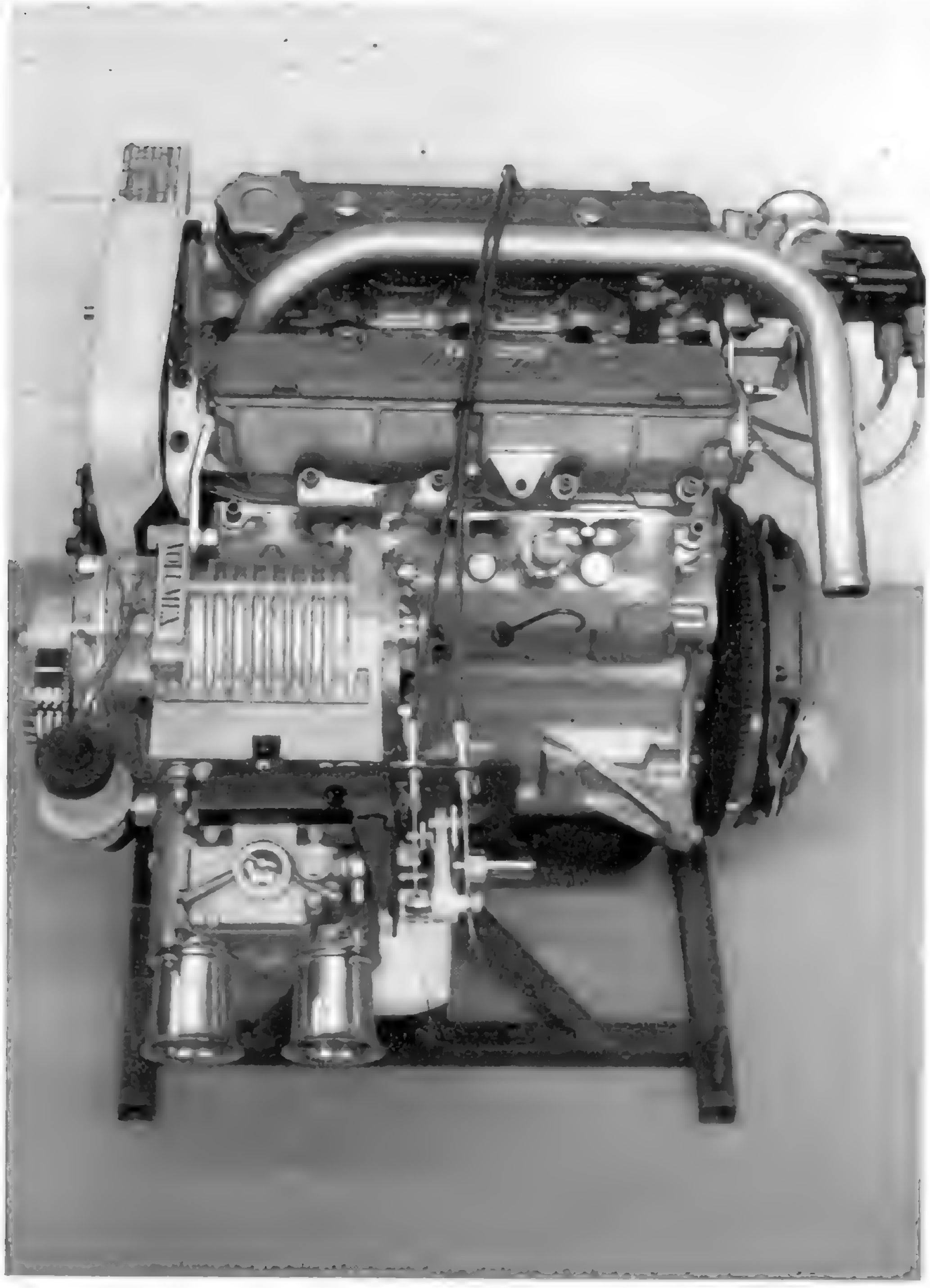
Notes: Hybrid inlet cam: The optimum type of profile incorporates low lift at TDC (around 3mm), very high peak lift (about 12mm actual) timing around 26/66 to 34/74°. Low LATDC retains reasonable torque at low speed, the high peak lift (with a medium lift integral – similar to GC IIB) gives prodigious torque.
(Oversize bearings and a line-bored cam box will be needed for such a cam due to the large base circle.)
Remapping of the fuelling EPROM is vital on fuel-injected units.



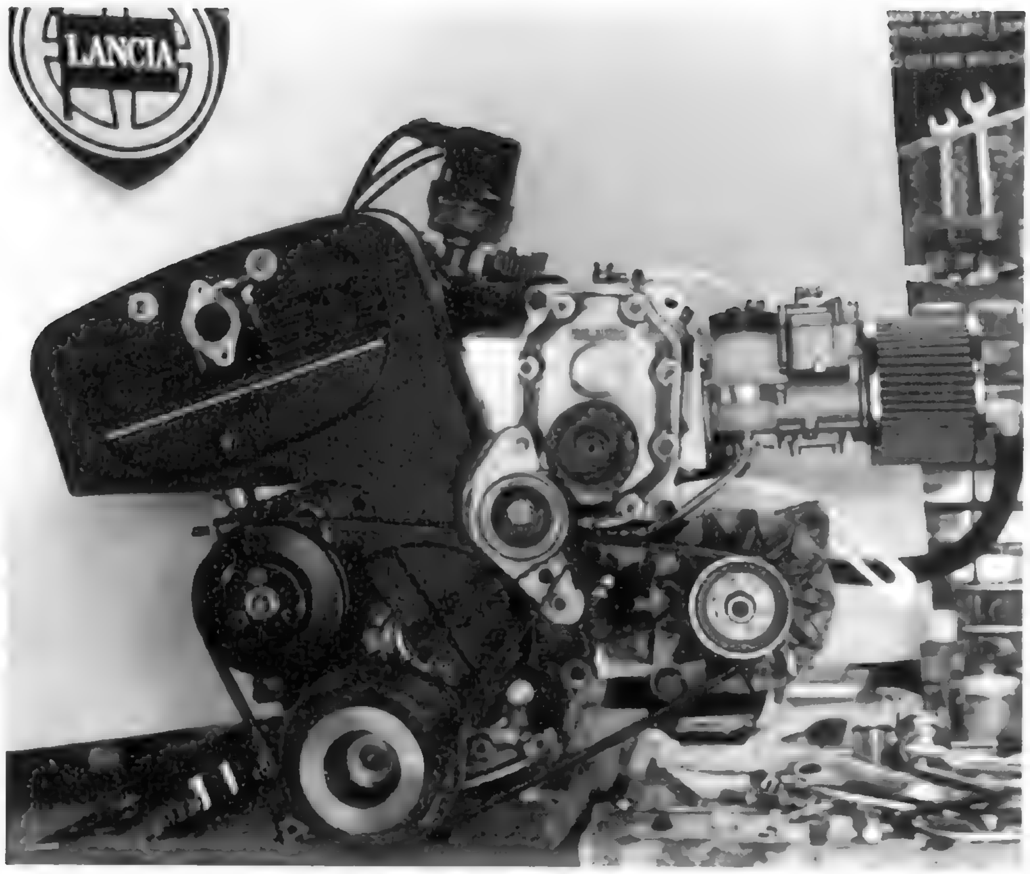
12/3: GC Volumex fitted with 40 DCNF. An easy conversion requiring modest alterations to throttle linkage. Engine was installed in Gerry Hawkridge's Stratos replica, superbly installed by Guy Mayers and featured in Feb '94 Kit Cars International magazine. Note direct-drive blower belt conversion made from 131 tensioner. A 1" belt should really be used.



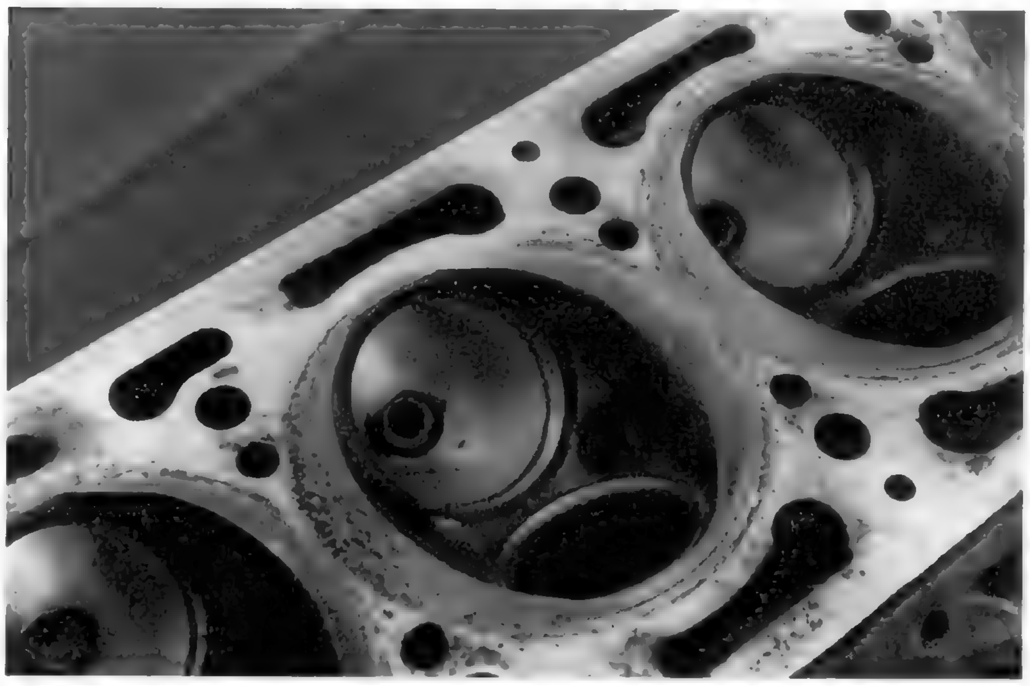
12/4: Volumex unit; twin-lobe Roots blower. Speed can be safely raised 7% by change of crank front pulley (see above). Essential to check oil reservoir for gears every 300–500 miles.



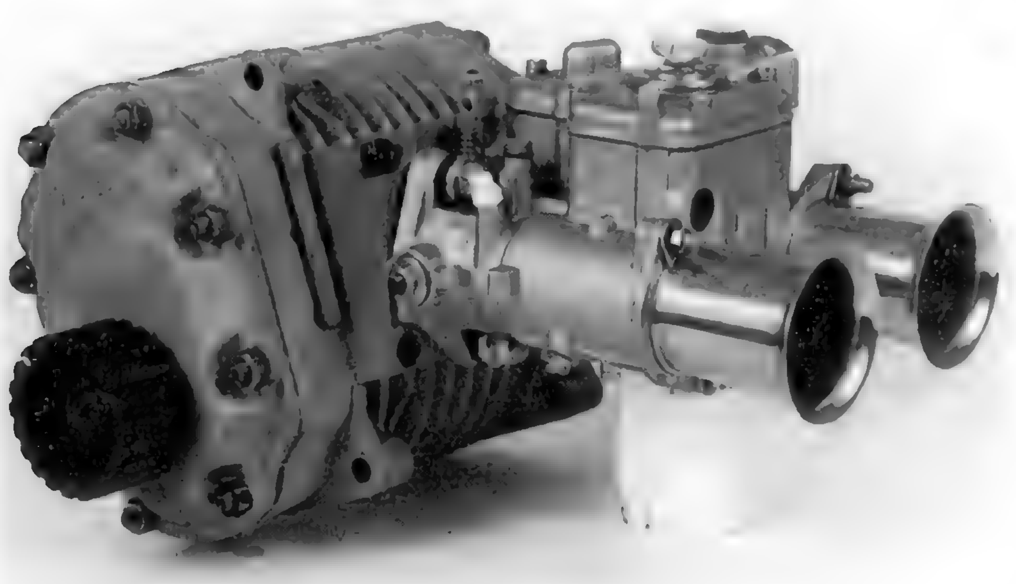
12/5: GC engine for Jerry Smith. Blueprinted, 7.5:1 CR: 44in valves, ported. Sidedraught 45 DCOE, Sachs Gp N clutch, direct-drive blower conversion. Estimated output 180bhp, 165lbf ft.



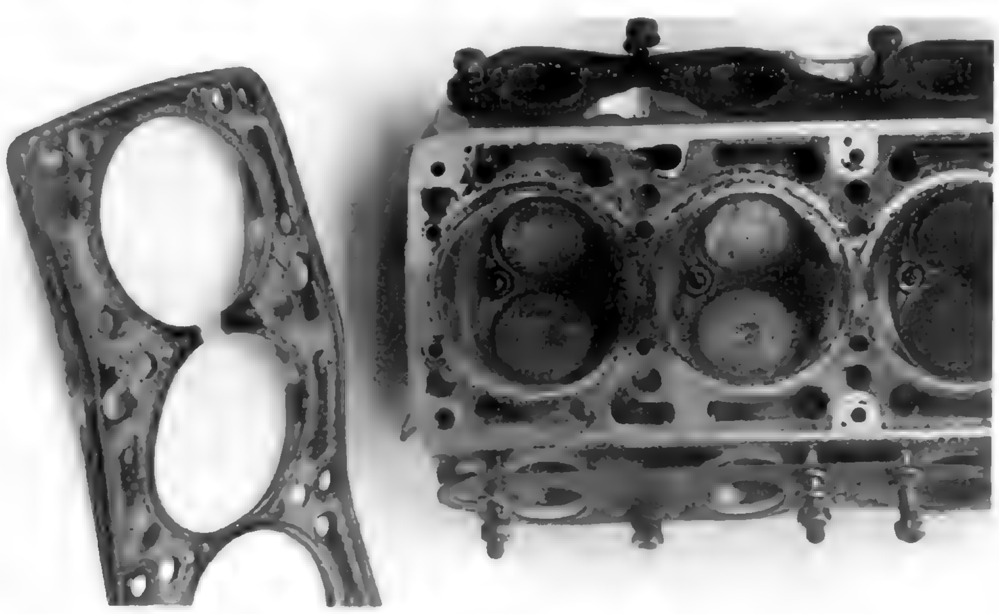
12/6: Neatly executed conversion (note adaptor plates either side of blower unit) allows Volumex to be fitted to Monte Carlo engine. Unit gave 175bhp @ 7000rpm. (Photo Tom McGaffigan)



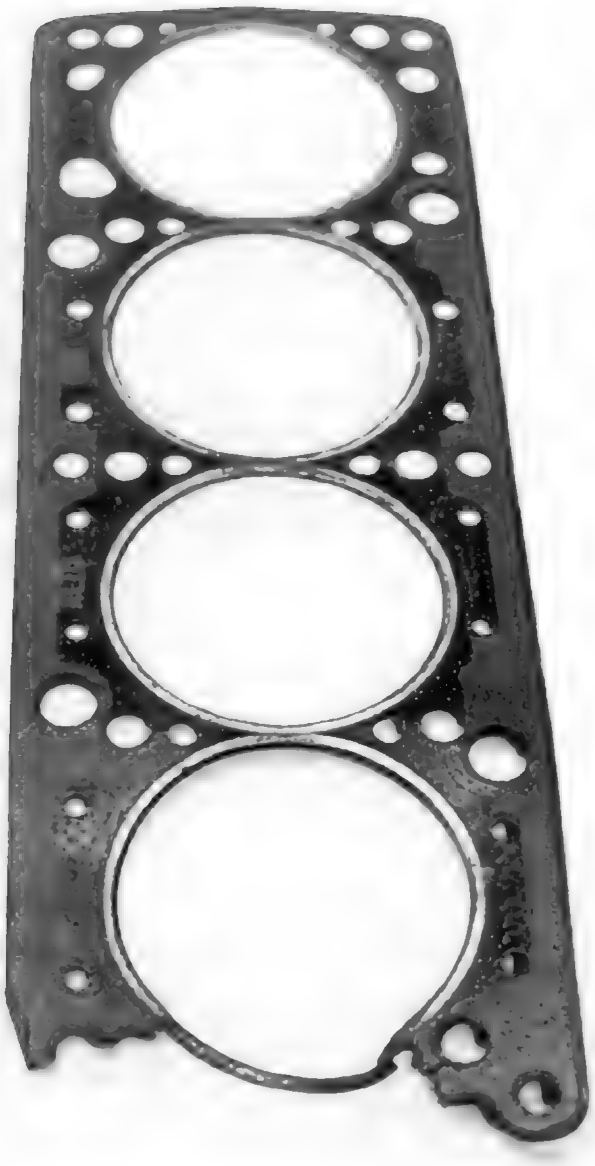
12/8: Severe detonation damage on this 1600 Delta HF turbo was caused by use of a non-turbo ignition system. Turbos require advance reduction from peak torque up. Micro Dynamics turbo ignition system cured problem. Head (fully ported etc – not GCT) was junk.



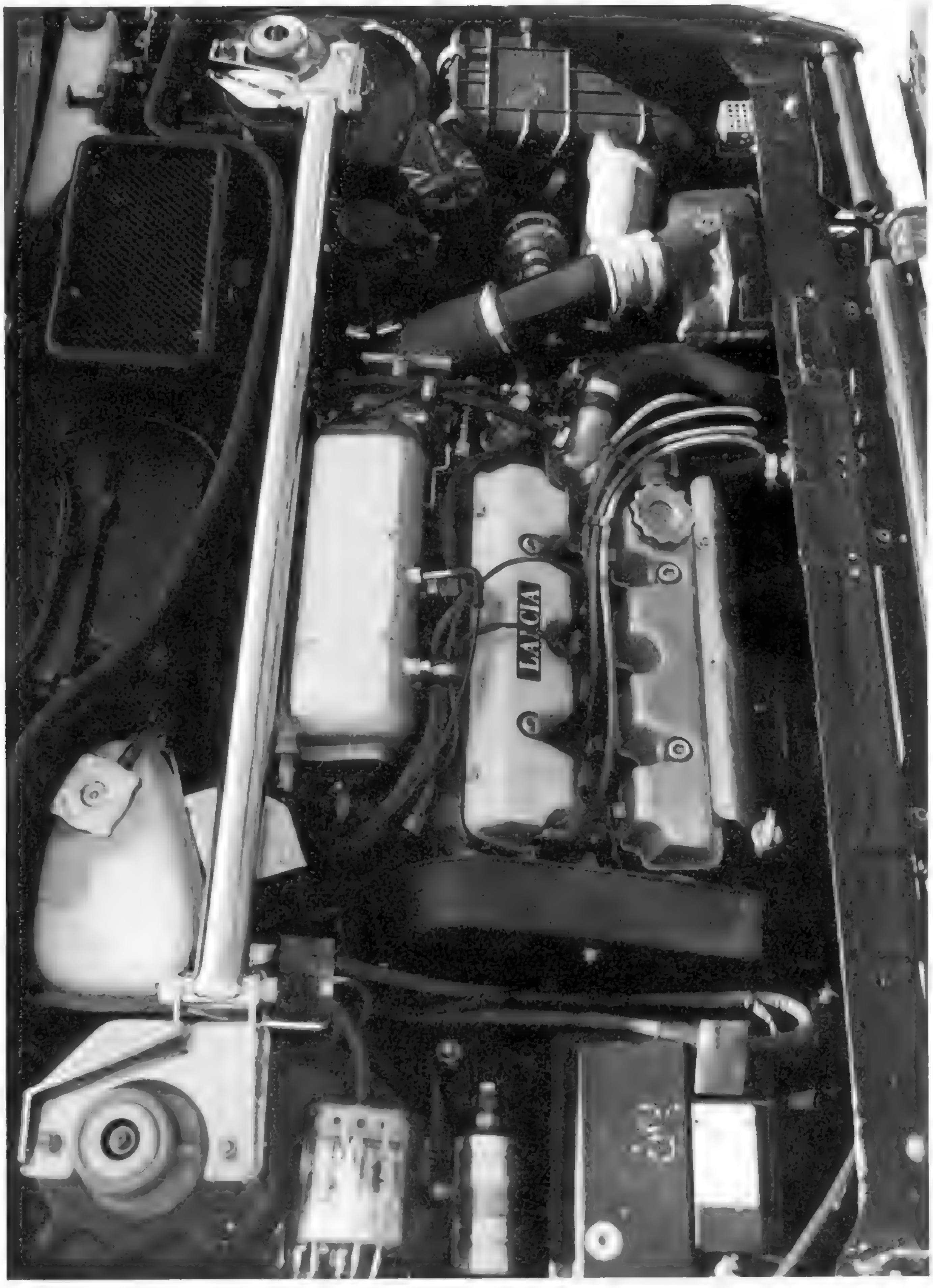
12/7: Lightweight GC adaptor manifold allows 45 DCOE to be bolted directly to Volumex unit. This conversion 'bolts on' about 25bhp and dispenses with need for standard manifold and its hosework. Add fuel pump (Facet Silver Top is ideal up to around 215bhp), blower vent pump, oil reservoir, and unit is ready to run. Standard inlet manifold suffers from 'pooling' of fuel (leading to hot starting problems – hence complicated fuel network of standard item) and adverse pressure wave effects – it is impossible to get more than about 180bhp from standard manifold (whatever carburettor is used).



12/9: A Volumex gasket should have been used! 131 (late) gasket blew under pressure (standard Volumex produces 152lbf ft torque). Head was damaged between 3 and 4 cylinders but welded up OK. Fitted by 'expert' who should have known better.



12/10: Gasket from Delta HF turbo on previous page.



12/11, 12/12: Right and above right, a 'Wolf in sheep's clothing', '96-style. This 'Integrale' is actually a highly modified Delta 1600 HF ie and is owned by Kelly Harris. Kelly, a toolmaker with CAV in North Kent, carried out all suspension, brake and body mods himself and built-up the GC-prepared engine. With a hybrid T25 turbo on 20lbf/in^2 boost, the car produced 223bhp @ 6686rpm and 208lbf ft torque at 4000rpm at Superchips rolling-road. Torque is 200lbf ft-plus from 3750 right through to 5500rpm! Road speeds (rolling-road figures at 7850rpm) are: 1st 45mph, 2nd 79mph and 3rd 119mph. Engine spec includes 44/36 valves, 105 TC cams, ported/blueprinted, fully heat-treated crank, 16v oil cooler, intercooler and radiator, with evolution Integrale water spray to intercooler (operates at 18lbf/in^2 boost).

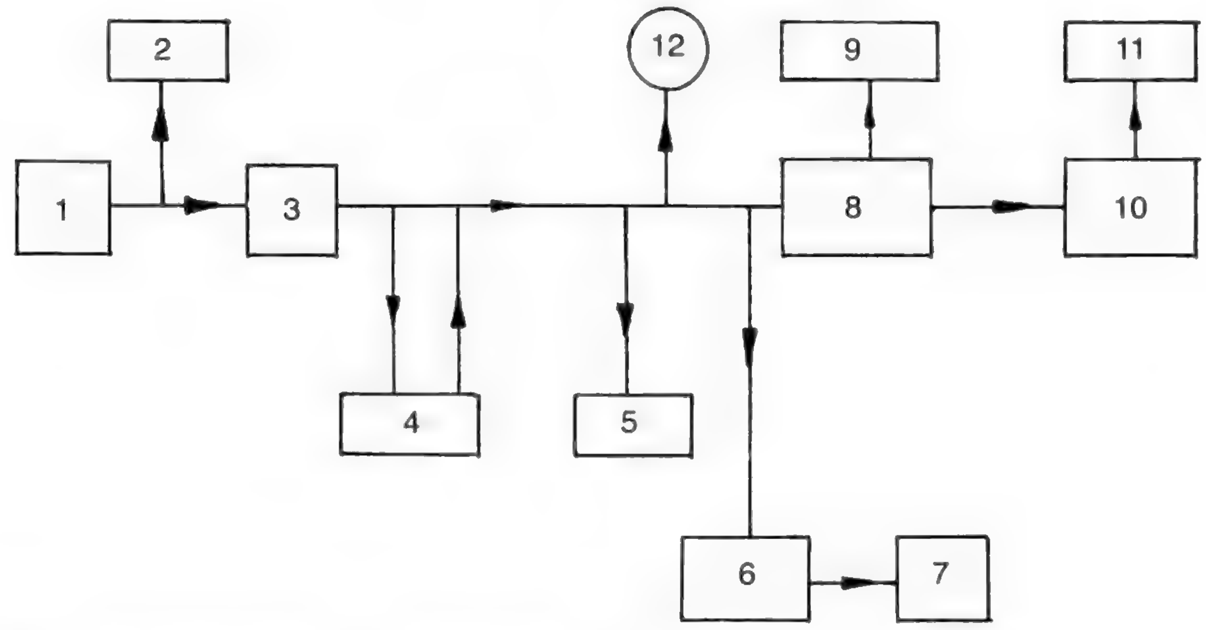
CHAPTER 13

LUBRICATION AND COOLING

LUBRICATION

Analysis of standard systems

All Fiat/Lancia TCs are equipped with wet-sump lubrication, the oil being picked up by the pump and fed through the oil circuit as shown diagrammatically here (13/1):



13/1: Diagramatic layout – lubrication circuit

Key
1 Oil pump
2 Oil pump/distributor, driven gear and bush (early models only)
3 Oil filter
4 Oil cooler – thermostatically controlled (where fitted)
5 Auxiliary driveshaft (early models only)
6 Balance shaft bearings (late models only) camshaft bearings, valve train
7 Distributor (head mounted only)
8 Crank main bearings
9 Piston cooling sprays (turbo models from Delta 1600 ie only)
10 Big-end bearings
11 Bore sprays on rods (not late turbo models)
12 Turbo (where fitted)

Notes on layouts

Oil return from all components is by drain-down (gravity). It is important to note that on early 8v TCs the oil drains in the cam boxes are designed to provide 'oil bath' lubrication to the valve train and maintain a specific level of oil in the cam boxes. Therefore, cam boxes from, for example, the Beta should have the rear box oil drains modified if used in a vertical installation, otherwise the cam box will flood. Late, reversed-port models, post-Delta Turbo *ie*, are fitted with oil restrictor valves in the head to augment the oil feed to the turbo.

On block-mounted distributor models the driven gear (driving oil pump/ distributor) is fed with unfiltered oil direct from the pump. If these engines are dry-sumped, no provision for a special feed to this gear (to drive the distributor) is required as oil splash and vapour will suffice.

The number of drillings on the 2*l* main journal was reduced on later models to augment the oil feed to the turbo and piston cooling sprays. The Volumex crank has larger main journal drillings than the normally aspirated 2*l* 131 and Beta, to increase the oil feed to the rods. It has been found at GCT that this is not necessary if a normally aspirated engine is being converted to a supercharged application.

The small-end bush is lubricated by oil draining from the oil control ring; early (non-fully floating) pistons receive oil into the piston boss in the same way.

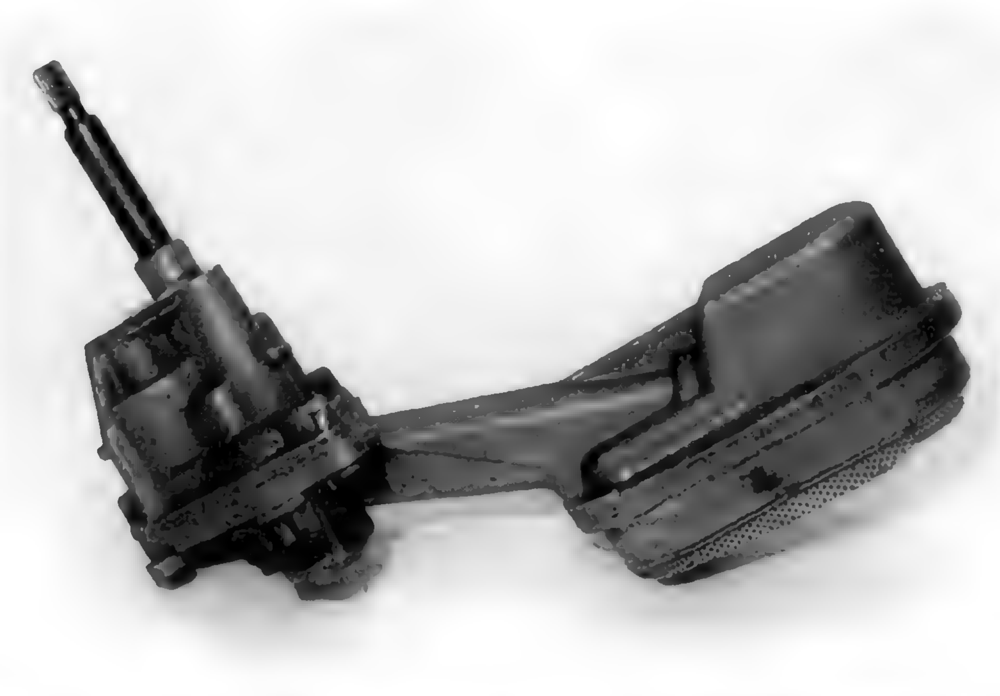
Early oil pumps are driven by a spline inside the driven gear and this mechanism is prone to wear. The 2*l* 131/132 oil pump shaft and gear were redesigned around 1981 to strengthen the shaft. Early pumps are all spur-external type; later models, post-Delta Turbo, adopted internal-gear crescent pumps driven off the crank nose. This design is extremely compact and gives far better flow at all engine speeds. All oil pumps incorporate relief valves, but spring poundage varies according to model, *eg* the oil pumps for the 105 TC and Delta Turbo (carb) are visually identical but the Turbo model has a heavier spring.

The output of the standard pumps in good condition has been found to be perfectly adequate on all models up to at least St III (wet sump) n/a, and 30lbf/in^2 boost on Turbo models. In fact, on a grasstrack 2*l* Fiat developing 175bhp @ 7000rpm, GCT used the 2*l* Beta pump (standard type) which has smaller gears than the 2*l* Fiat (albeit with a heavier relief valve spring), because its lower profile gave the chassis clearance required. As a general rule, under load (*ie* from 3500rpm-plus) approximately 10lbf/in^2 per 1000rpm is required. A TC may tick over with anything between 15–25lbf/in^2 depending on oil temperature and crank clearances.

Oil pumps – useful data				
Engine	**Pump Part No**	**Driven gear Part No**	**Output at 6000rpm (hot)**	**Notes**
131 1600	4379251	4274698	43–57lbf/in²	
105/130 TC	7541295	7541261	50–75lbf/in²	Same upper body as late 131 2*l* (7541173)
124 1800	4199918	as 131 1600	as 131 1600	
Argenta	5933760	originally used	as 130 TC	Short pick-up. Use late early 131 2*l* type (7541173) pump assembly on replacement with 130 TC gear
124 1608	4199918	as 131 1600	as 131 1600	
Delta HF1600 *ie*	7717403	crank driven	50–75lbf/in²	Crank nose driven
Delta HF1600 (carb)	7541296	early 2*l* 131	as 131 1600	Driven off aux d/s
131 2*l*	7541173	as 130 TC	as 130 TC	Late with redesigned spline: late gear shown must be used
Volumex	7032541	as 131 1600	66–85lbf/in²	Upper body interchangeable with 82334025
Beta 2*l*	82334025	as 131 1600	66–85lbf/in²	
Beta 1600 (1585)	82317492	as 131 1600	66–85lbf/in²	
Monte Carlo	91101748	originally used early 131 2*l*	as 130 TC	No longer available, use 131 2*l* upper body plus 7541173 gear

Critical checks		
1	Splines	
	Clearances	
2	Rotor to casing	4–7thou"
3	End float	1–4thou"
4	Backlash	6thou" approx (not critical, new pumps often have more)

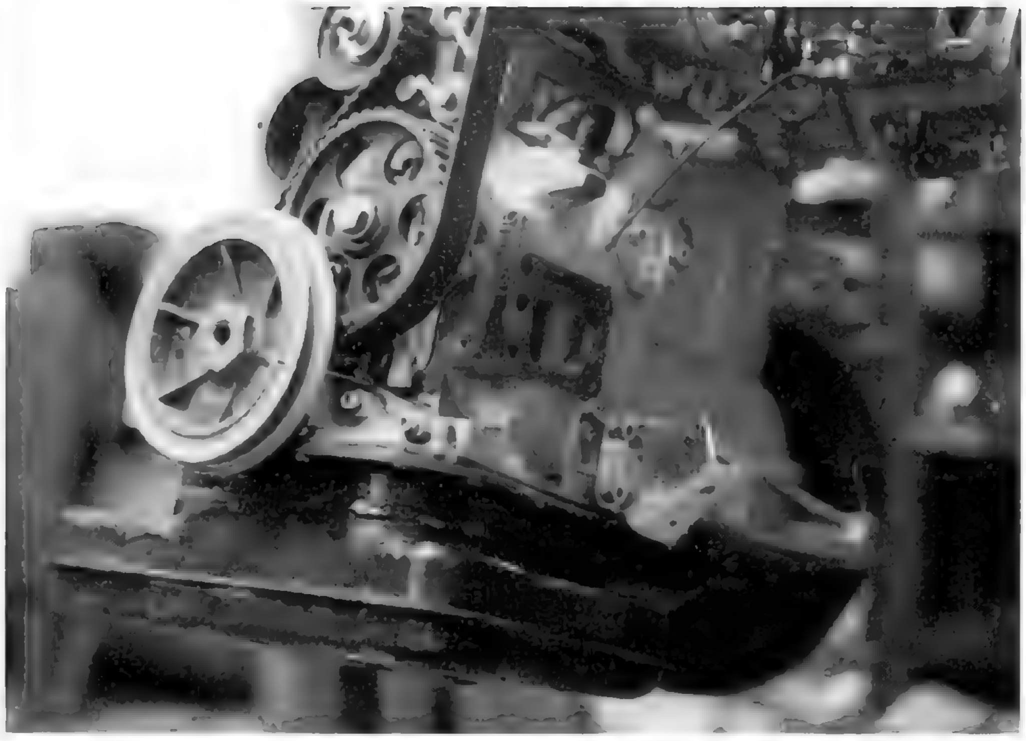
13/2: Lancia Beta 1585 oil pump and driven (skew) gear. Always replace gear if replacing pump as internal spline will be worn. This 2l pump has taller rotors/body but is still only same size as Fiat 1800 (but with a much heavier relief spring). If you want to know history of an engine – have a look inside the oil pump strainer!



13/3: Volumex oil pump. Cavity above pickup is designed to prevent oil starvation; the pump delivery is so high, however, it won't prevent it for long! 2l Beta pump has external relief valve, 1585 is internal. Note stress-raiser at sharp change of section on shaft.

13/4: Pickups. Left rear – Delta/105/130 TC (fits 131 2l upper body) flange-mounted pickup, useful for rear-well conversions.
Left front – Monte Carlo (also accepts 2l 131 pump).
Right rear – Volumex (uses standard 2l Beta pump body with different pickup).
Right front – 131 2l (note that 124 1800 pump will not interchange, unlike 132 1800, as mounting points are different).

13/5: Various oil ancillaries. Left – 124 Sport oil filter housing with two tappings for gauges.
Centre – 131 2l oil filter housing (one tapping only).
Right – Beta type, which also carries alternator, two tappings.
Front left – breather separator unit from early model TC.
Front right – Delta oil filter take-off plate with unions to oil cooler and remote filter.



13/6: Argenta engine. Crude pressed-steel 'big wing' sump would have been very effective if fitted with trap door baffles on boxes. Unfortunately it isn't and it's not possible to fit them due to design of pickup skirt! Pump has special short pickup designed to operate approx 1cm above sump base.

Standard sump analysis

Type	Design feature	Protection against Starvation	Protection against Aeration	Notes
124/131	front well, small static baffle, some models have small baffle around pickup	reasonable	negligible	Suitable for fast road only. Vulnerable pickup/base
Argenta	as 131, lower profile with boxes (no trapdoors)	quite good	negligible	Fast road
105/130 TC Delta/Prisma 1600 (n/a and turbo)	deep full-length transverse sump, vertical installation, cruciform static baffles	very good	nil	Up to St II. Croma sump also quite good
Monte Carlo	deep full-length transverse sump, 20° rear tilt. Twin bulkheads and trapdoor baffles	excellent	very good	Up to St III
Beta	shallow ¾-length poor transverse sump, 20° rear tilt, centre bulkhead (some models have perforated static baffle around pickup)	nil	nil	Not suitable for anything except very mildly tuned road use
Integrale	deep full-length alloy sump with steel base. 20° forward transverse, modest static baffles and windage tray	quite good	reasonable	Suitable for Gp N only (for Gp A use Accusump)

Race sump design

The purpose of a sump is to contain the engine oil. In order to lubricate the engine, the oil needs to be in constant supply and at the right temperature.

First problem: if the car is swinging around and the 'g' forces are applied to the oil, how to ensure it stays near the oil pump pickup. During the cornering condition and under braking and acceleration, of course, the oil tends to climb up the walls of the sump and crankcase, away from the pickup, *ie* the pump runs dry and pumps air (or crankcase gas).

Second problem: the oil (with certain types of sump) can be in constant contact (so to speak) with the crankshaft. The crank has the same effect as an egg whisk, so it is easy to appreciate what happens when a crankshaft rotating at anything between 5000rpm and 9500rpm (a full-race 1585 can develop peak power at over 8500rpm) is applied to the oil. The technical term is aeration, a mild-sounding term for something which, as far as engines are concerned, is a killer.

Third problem: temperature. Excessive oil temperature degrades the molecular structure of the oil (it goes thin and can weaken), leading to bearing and seal damage and, since part of the job of the oil is to cool the engine (about 20% of the overall heat of the engine is dispersed through the oil), it can't do that either.

The ideal solution would be if manufacturers would fit excellent sumps to their production cars, but unfortunately in the majority of cases the sump is hung onto the engine as an afterthought (the Lancia Beta is a prime example). Some of the design features needed are:

1 The oil must be contained around the pickup to keep it in position under the forces of left and right turns, acceleration and deceleration.

2 The oil must be separated from the crank and rods to cut down aeration and prevent oil contacting the crank (especially in the rally 'yump' mode!).
3 The sump must have adequate capacity and heat dispersal or must be supplemented by a secondary system to cool the oil.

The layouts of FWD and RWD sumps are, quite properly, very different. In part, the design is dictated by the need for adequate oil capacity to prevent over-heating; in the case of the Beta and 130 types the sump necessarily extends the full length of the engine, whereas the 124 and 131 type incorporates a deep well at the front.

Clearly, in the case of the FWD sump, if the sump us not well baffled there will be a serious oil surge problem under braking and acceleration due to the shallow depth and the limited amount of oil adjacent to the pickup for F-R transfer: the oil will simply climb the walls of the sump and crankcase.

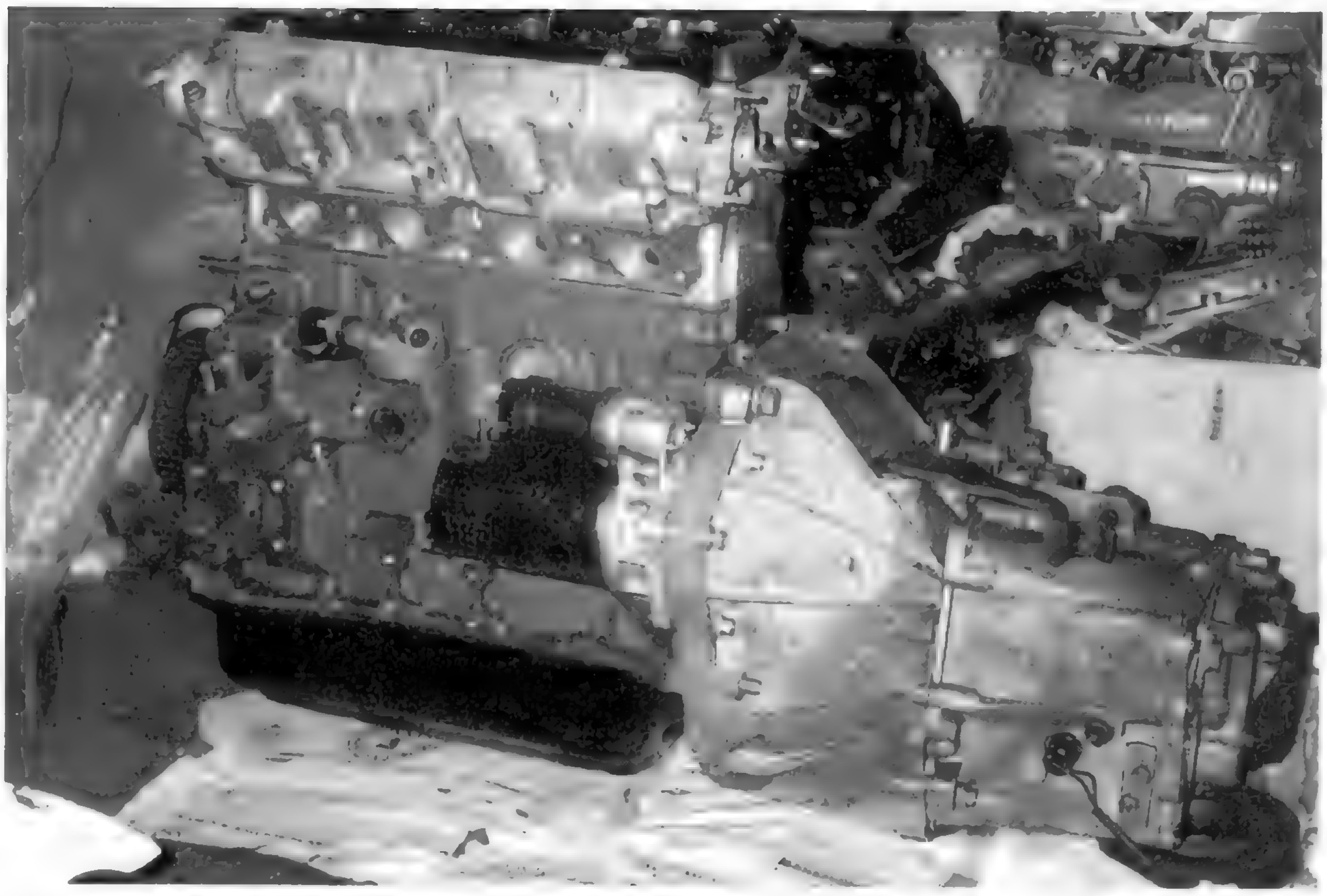
The problem is worse during cornering, where the oil achieves high velocity under cornering 'g' forces and sweeps straight past the pickup if not controlled by baffles. The well-type sump scores better due to its depth and smaller contained area, but during acceleration the oil tends to surge backwards, and from this point of view the Ford 2*l* 'rear well' design is superior. During braking, the bearing load is lower anyway (except on 4WD turbo cars when left-foot braking is used) and forward surge is not so significant a problem – unless it empties the well completely.

Over the years, much time has been devoted at GCT to improving certain Fiat models. It is easy to weld strips of metal in place to try to contain oil around the pickup, but unless a windage tray is placed over them, under hard 'g' the oil will simply climb over the top! If the windage tray seals the baffles completely, the oil cannot get in or out.

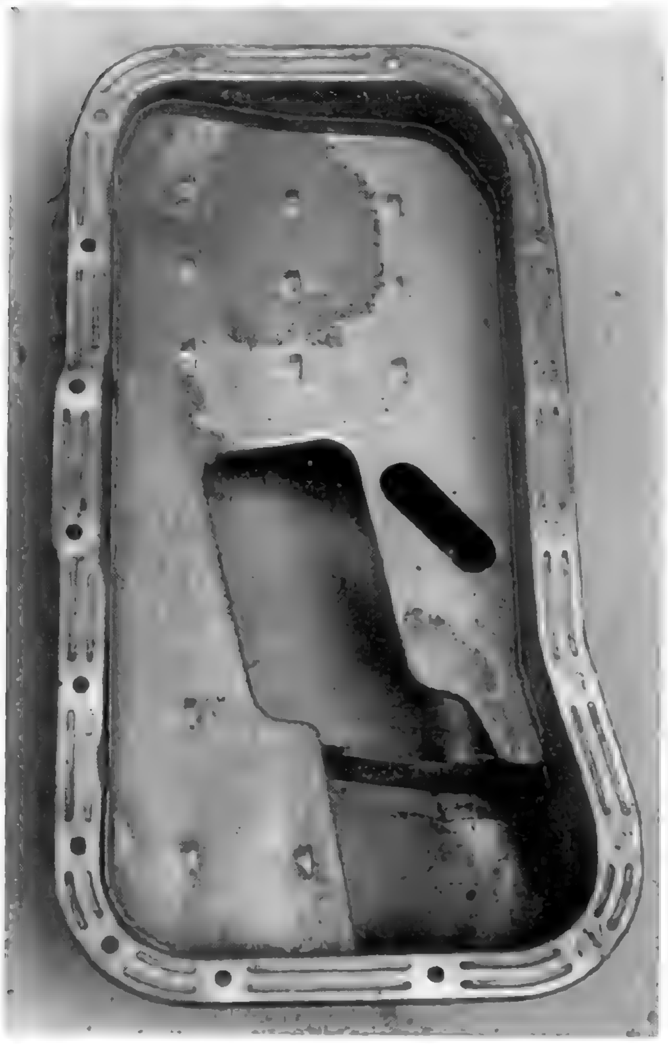
Now realizing the shortcomings of the Beta sump (nearly all the Beta engines stripped at GCT have damaged bearings), Fiat introduced a new design with the Delta, 105/130 TC etc, which has a rather clever cruciform design whereby with a very small centrally located pickup, whichever direction the oil moves it always flows across the pickup. Sadly, this will not fit the cars with a Beta-type crossmember and the pickup cannot be married easily to the Beta pump (it needs welding). The designer unfortunately didn't fit a windage tray, though of course it would have worked extremely well.

Another way to contain the oil is to fit static skirts around the pickup and fit one-way trapdoor baffles in them so that the oil can flow in towards the pickup but not away from it; it follows that the skirts must be covered by a windage tray which will stop the oil climbing over the tops of the skirts, keep the oil away from the crank, and allow oil to drain back down from the engine into the sump.

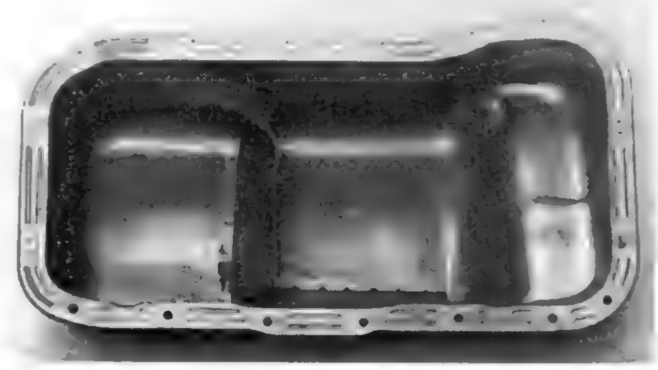
The approach with well-type (RWD) sumps is rather different. The object is to reduce the tendency of the oil to flow backwards to the rear of the sump, and at the same time provide 'transfer boxes' on the sides of the sump. By doing this the ground clearance can be increased. The transfer boxes should be designed to fill slowly but empty quickly, so that oil tends to be retained around the pickup as the box fills. For example, on a RH turn the LH box fills slowly, but the RH box discharges quickly over the pickup to prevent oil starvation.



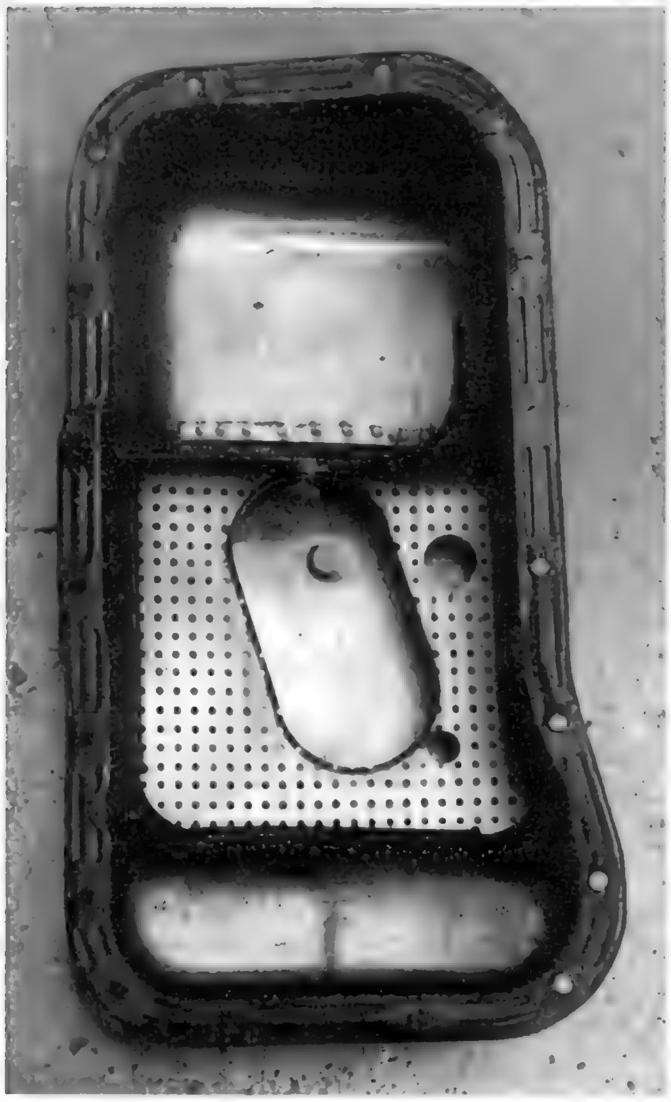
13/8: Delta 1600 sump fitted on Peter Gerrish's unusual 2l Delta. Note full-length design for vertical engine. Delta gearbox (turbo) has 1600 GT input shaft to mate to 2l clutch. Peter was one of first to try late (reversed-port) head design; ports visible are exhaust. Note large stud to left of scavenge unit – block is Monte Carlo. Delta 1600 oil pump and driven gear fit straight on Beta 2l block (pump has same output as 131 2l).



13/7: A classic design was sump fitted to Monte Carlo. Team who thought this up really knew what they were at; windage tray, central pickup, twin bulkheads with twin trapdoor baffles. Insofar as any 'wet' sump goes, this design represented ultimate idea. Once again, though, will not accept subframe instrusion of Beta design and is not compatible with Beta pump, though Monte Carlo pump will fit Beta block. However, pan is no longer available new and is highly sought after secondhand. QED.



13/9: Early Beta sump. 1600-2l types are different depth but designs are equally hopeless for anything other than mild road use.

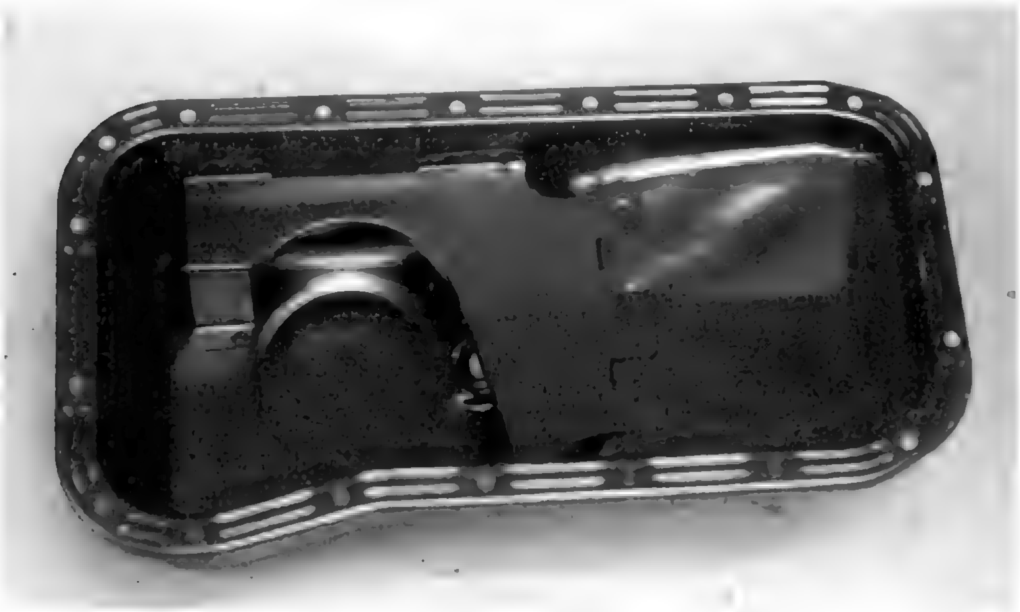


13/10: Later model Beta pan incorporates modest baffle plate around pickup. If sump is being fully baffled, as described later, this is worth retaining.

13/11: In order to turn Beta sump into something respectable it is first necessary to rip out standard internals; do this by part-drilling accessible spot welds and then attacking standard skirt with big hammer and sharp bolster chisel. Crude as it may seem this usually gets unwanted bits out without too much drama, although one or two spot welds have to be welded-up. Bulkhead fitted with one-way trapdoor can then be welded in place and full-length windage tray fitted over top. Windage tray is slotted to allow drain-down, and has further static baffle around pickup itself clearing base of sump by about 1cm. It's not really practical to fit any more inside Beta sump, and they have slight disadvantage that they work better on RH corners than left. But they do work substantially better than standard unit and, of course, greatly reduce aeration effect because windage tray keeps oil away from crank assembly.

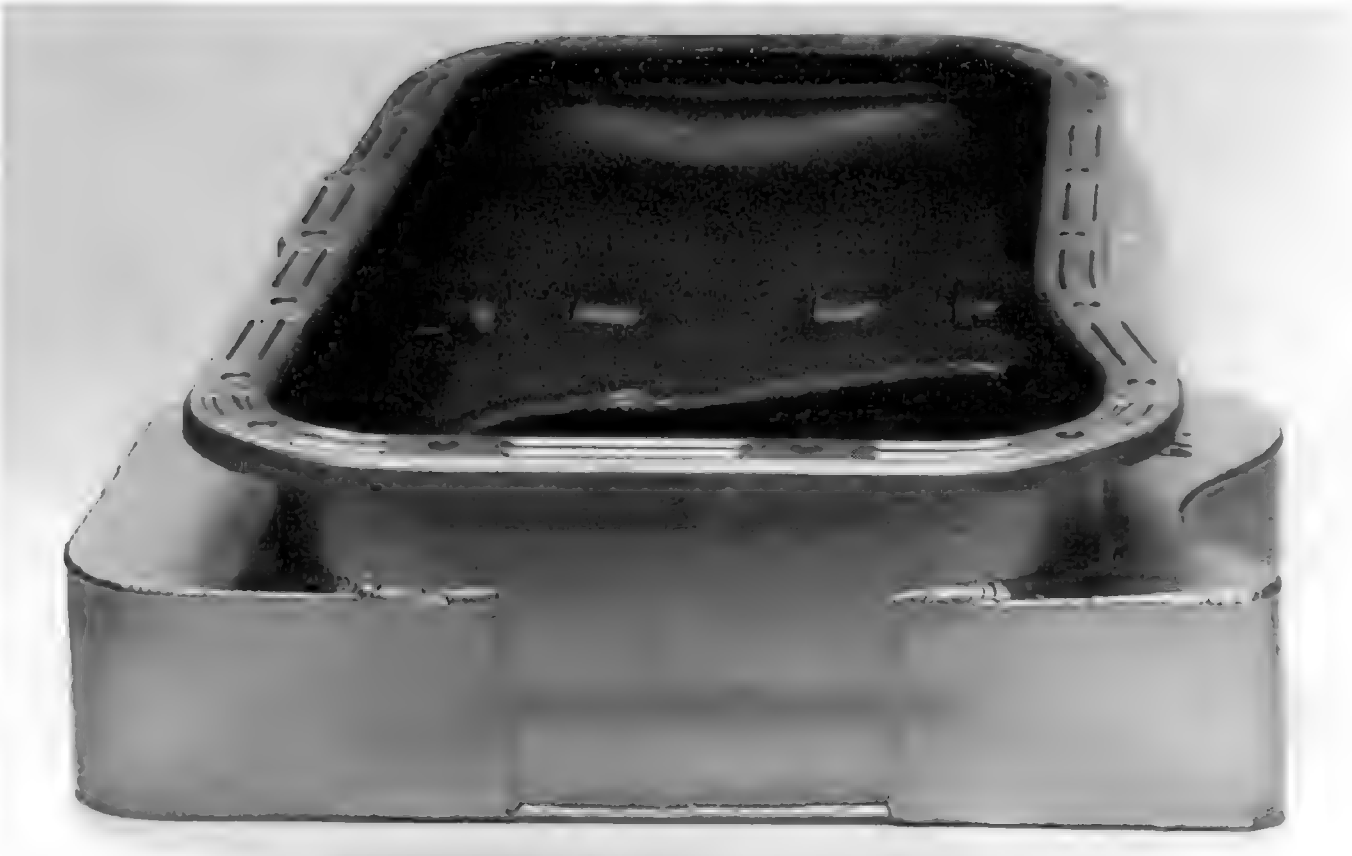
13/12: Fully baffled Beta sump. Spot welds just left of centre on windage tray show position of bulkhead plate (dotted line). This design will also accept the Volumex pump. Note clearance along sides of tray to encourage drain-down.

13/13: Standard (if well battered!) 2l 131 sump compared with steel works 1800 124 type. Note scallop in LH transfer box on works sump to allow for block-mounted filter. This design gives poor performance on LH turns due to small box oil capacity.

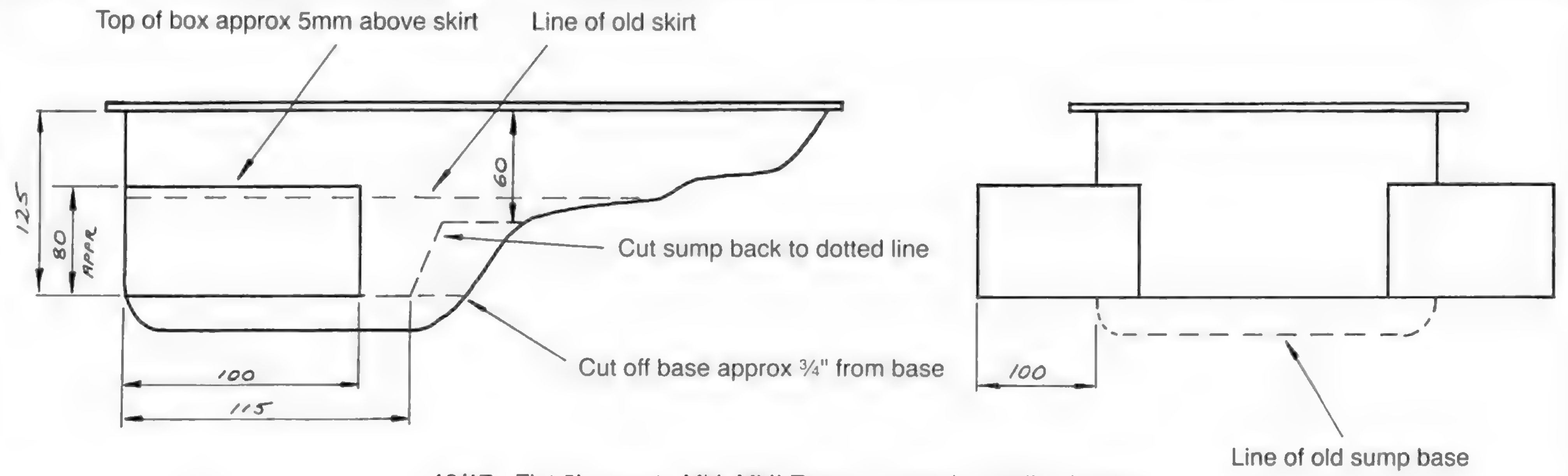


13/14: Sump is 124 1800. Note lack of recess in rear of sump for steering rack and no offset on sump bolts on side (unlike 2l). Moderately effective design for mild road use only. Very exposed pickup.

13/15: Welding sump base in position of 131 2l big-wing sump prior to construction of boxes. Vent holes and trapdoor are already in place. MIG is shown – OK for tacking; TIG is far superior.



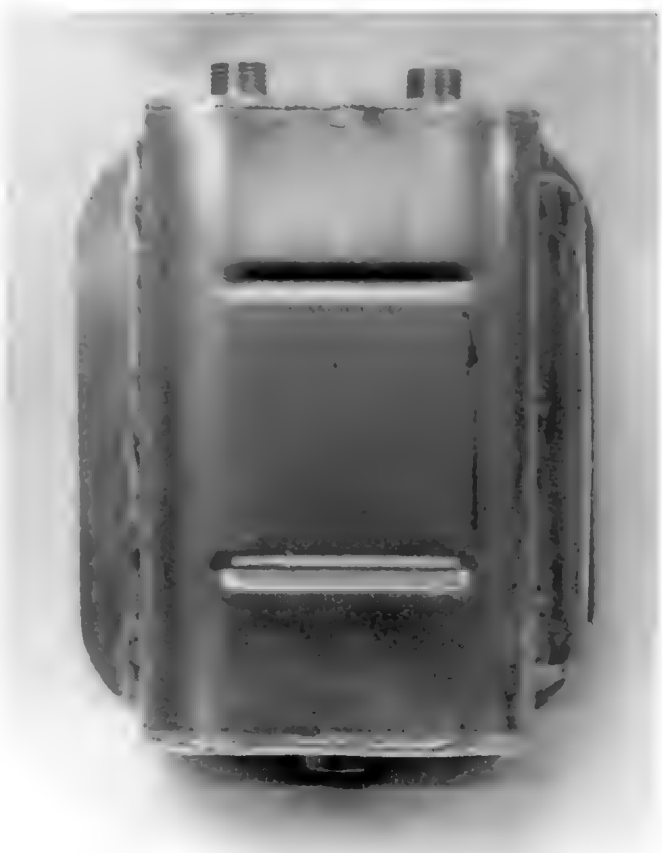
13/16: Early Guy Croft 'works replica' big-wing sump. 'Scallop' for filter impaired performance on LH turns. Sides of boxes are single strip, tacked and formed to exact shape. Note height of boxes relative to internal skirt.

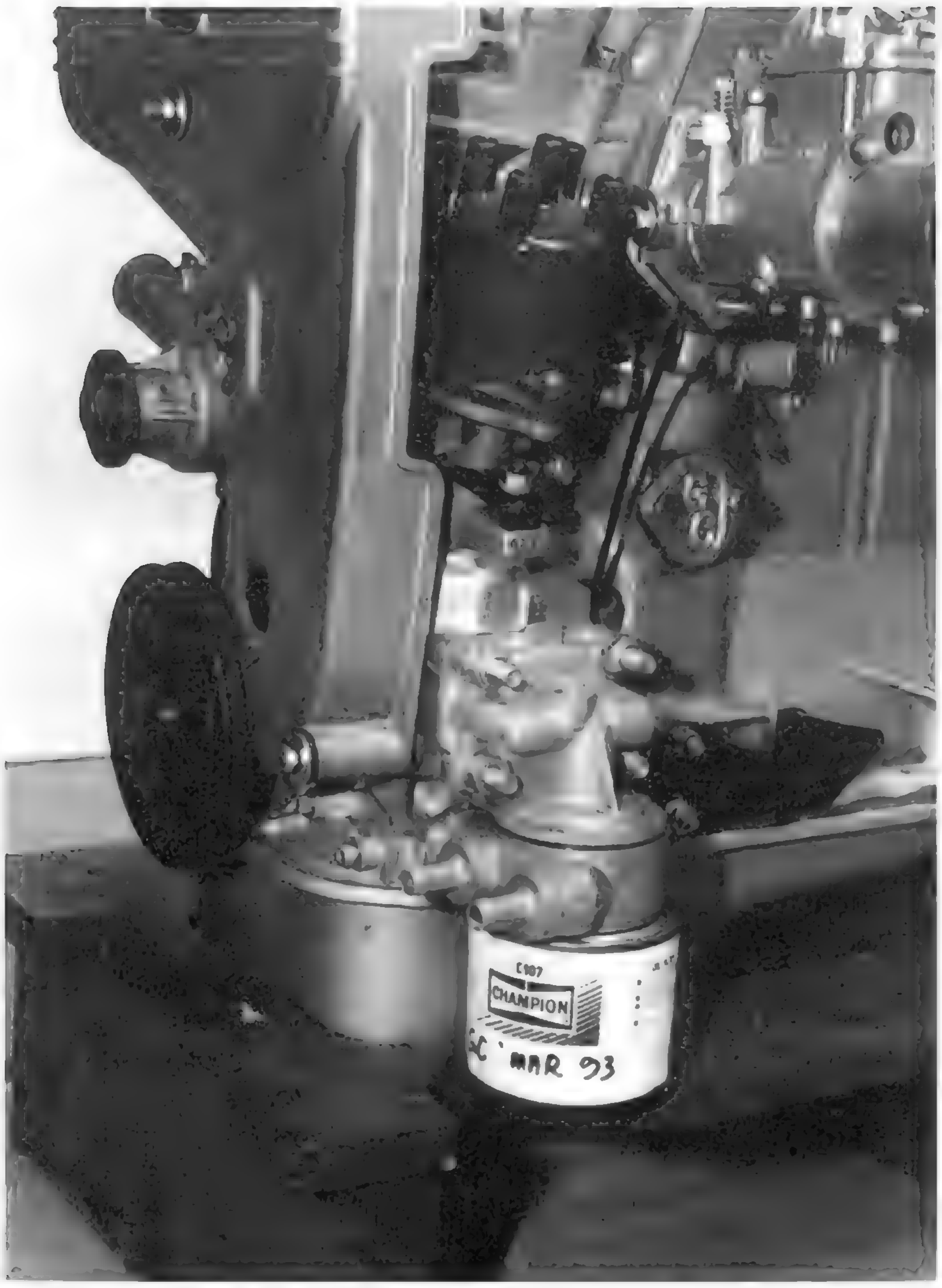


13/17: Fiat 2l sump to MkI–MkII Escort conversion outline layout.

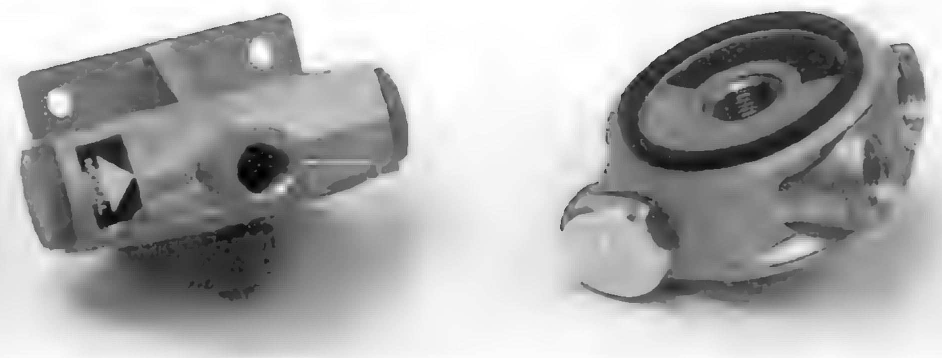
13/18: Latest-pattern GC 'full form' big-wing sump, extensively proven in NHRA racing. Skirt hides one-way trapdoor baffles on each side. Holes drilled inside sump are to help filling of box and allow it to vent as oil level rises. Base is 2mm, boxes 1.5mm. Base is approx ¾" less deep than standard on 2l version, 1600 (1585) can lose about 1".

13/19: Usual procedure with breather on competition cars is to vent to a catchtank. This model is inexpensively produced by Brise Alloy Fabrications of Dartford: incorporates ½BSP in-out unions, sight gauge and drain plug. Tank collects liquid oil and allows gas to escape and is an RAC race-legal requirement on all competition cars. Brise are well known for their excellent fuel tanks, dry-sump tanks and other alloy accessories.



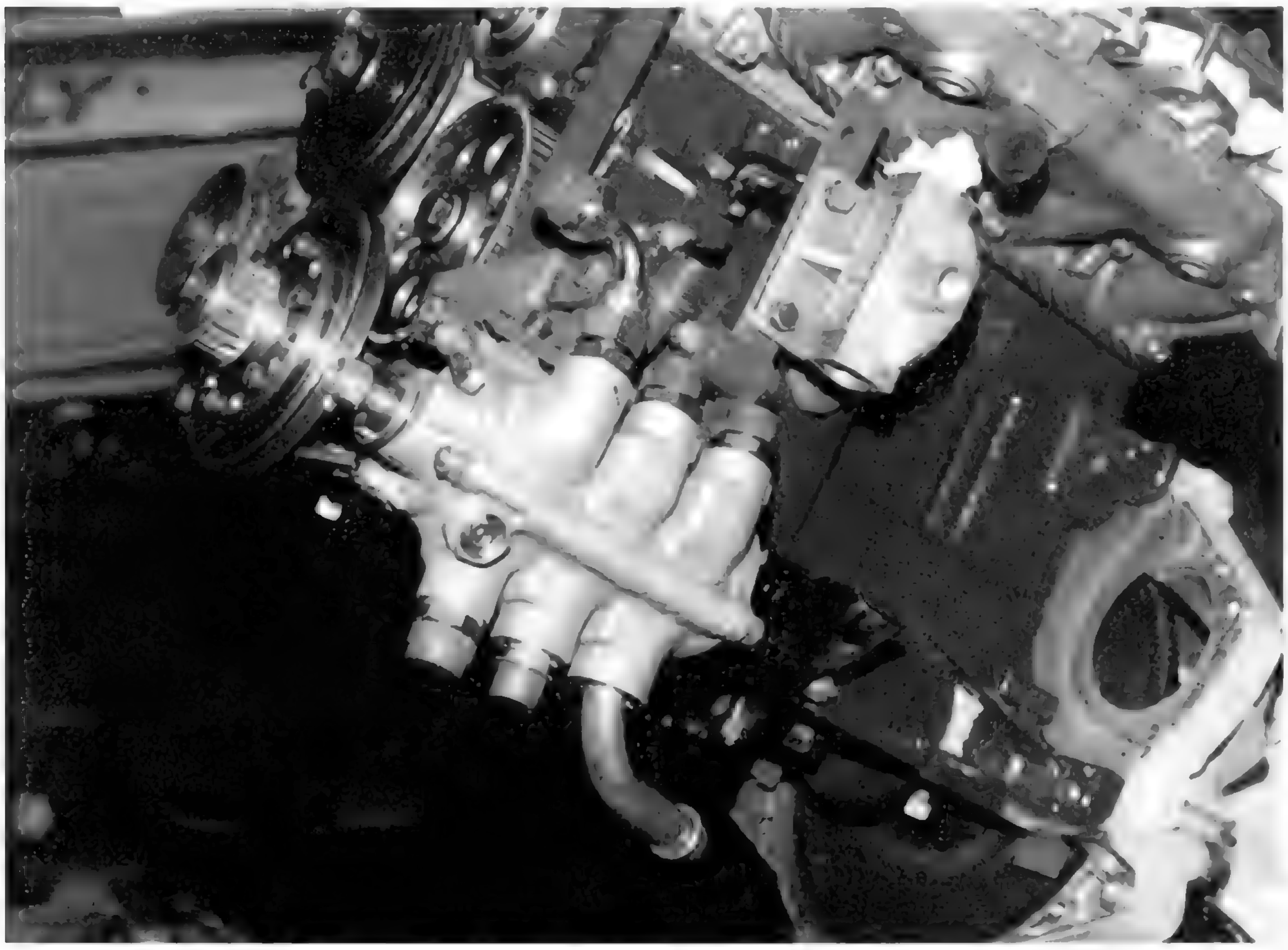


13/20: This particular thermostatic sandwich block showed a tendency to stick shut on the Duckhams QXR Monte Carlo and is no longer used by GCT. Engine is 125 Samantha.



13/21: Top-quality oil accessories from Mocal (Think Automotive). Left – remote filter head (available in variety of configurations). Has tapping for oil pressure gauge. Right – thermostatic sandwich block. Device allows full-flow through oil cooler at 84°C. Also available as non-thermostatic model.

13/22: Take-off plate (124/131) designed for use with 'full form' big-wing race sump, has ½BSP in/out unions to remote filter and ⅛NPT tapping for low-pressure warning switch.



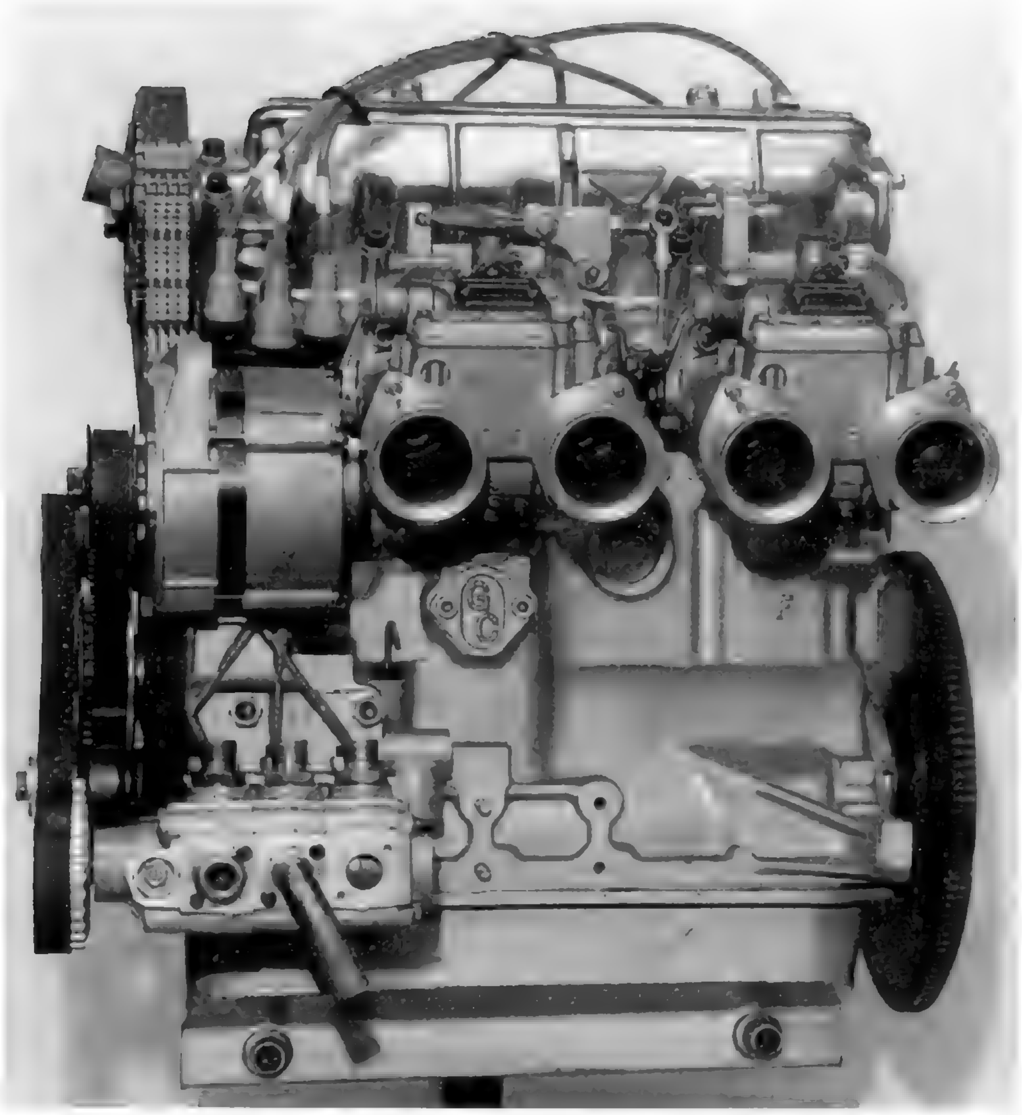
13/23: Series 1 Titan 3-stage dry-sump pump fitted to author's 124 CSA during '85/'86. Crank driven on 50% speed reduction.

Dry-sump lubrication

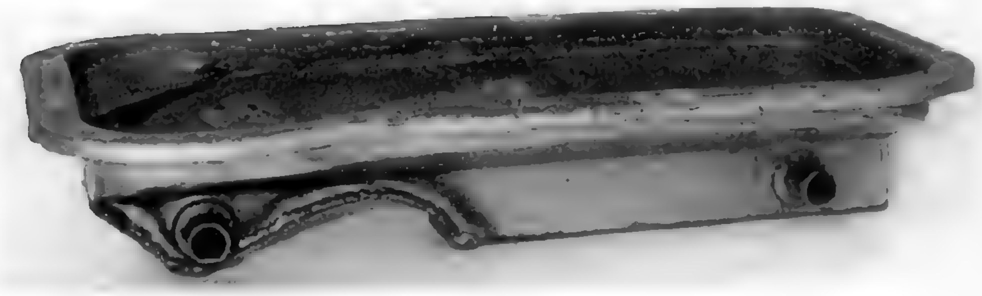
A dry-sump system differs from a wet-sump in that instead of oil being contained in the sump it is stored in a remote tank and pumped to and from the engine by an external pump. The pump is driven from the crank or auxiliary driveshaft via a toothed belt and, in the case of the TC, GCT usually employ a three-stage design, with two scavenge sections and one feed stage.

The scavenge pumps draw oil and crankcase gas from the pump and return it to the dry-sump tank, and the feed stage takes oil from the tank and pumps it to the main oil feed gallery. It is usual to put the oil cooler on the scavenge lines to reduce pressure loss and to pass the oil through the filter last of all, before it enters the engine. The dry-sump pan should incorporate a windage tray under which the oil is thrown by the crank effect, and gauze scavenge filters which prevent any serious foreign material damaging the pump.

A properly designed tank will incorporate baffles to de-aerate the oil, and this is one of the primary advantages of the

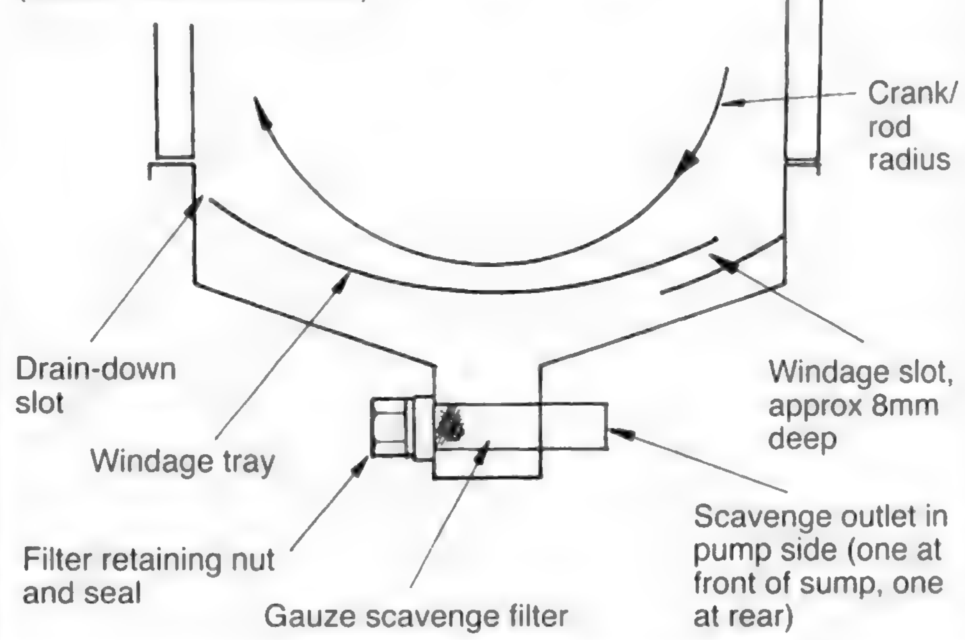


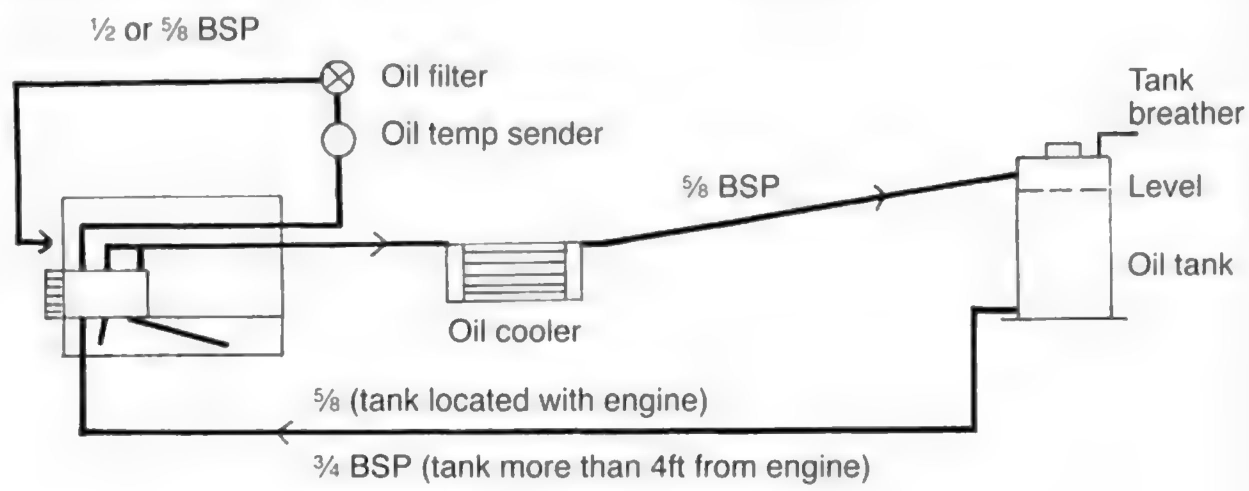
13/24: Guy Croft St IV Race Beta unit showing layout of mountings and drives of alternator and pump with dry-sump system. Alternator top bracket is not yet fitted. A very tight installation – originally built for 24-hour race, hence alternator. Sump pan is made by mating new base section to old Lancia sump.



13/25: Well, you can't be right all the time! Low-profile dry-sump pan for Tom Casey's Fiat Hot Rod didn't fit in car! Had to be locally modified. [Not the best welding in the world but it worked OK.] TIG rather than MIG welding is way to make sumps.

13/26: SCHEMATIC VIEW OF DRY-SUMP PAN (viewed from crank nose)





13/27: DIAGRAMATIC LAYOUT OF DRY-SUMP SYSTEM

system, the other, of course, being that oil starvation is totally eliminated. Clearly, the pump design must ensure that the capacity of the scavenge stages exceeds that of the pressure stage or the engine will rapidly fill with oil. As a general rule the pump should be driven at 50% crank speed.

Dry-sump belts are vulnerable to damage by stones. For rally or similar applications, Accusump is preferred (see later). Another weak point of the system is the vulnerability of the belt to damage when the oil is cold – an oil pre-heater is a useful accessory if 'instant start and race' is envisaged. A certain amount of oil tends to accumulate in the sump when the system is left standing for days – always check the tank level before start-up and if necessary evacuate the sump with an external drive to the pump, *eg* an electric drill, to avoid rod damage by hydraulic locking, or incorporate a shut-off valve in the feed line from the tank to the pressure stage. (Don't forget to open it before you start the engine!)

There is no hard and fast rule as to when dry-sump lubrication should be used. In general, when the horsepower exceeds 95bhp/litre and race regulations permit, it is a highly desirable option, especially if race-material bearings are not available (*eg* 1585 rods). Certainly high-output, HC race engines tend to produce more crankcase gas due to blow-by, and dry-sumping will eliminate this problem. Some engine builders claim a power increase from dry-sumping, but GCT have no data to substantiate this.

INSTALLATION HINTS (see layout diagram – 13/26, 13/27)

Locate the tank on the same level as the pump to reduce pumping loss to the feed stage; preferably in an airstream and well away from hot components, *eg* exhaust. Do not fill the tank above the level of the baffle plate. Always use 'swept' oil line connections, not 90° cranked fittings, to reduce pressure loss. A cranked fitting has as much pressure loss as a 6ft pipe.

Hose selection

Suction hose (sump to pump and tank to feed stage): Aeroquip steel-braided Teflon hose or steel internally braided rubber hose.

Pressure hose: Aeroquip rubber-braided (textile or steel).

Steel or textile externally braided rubber hoses should be nitrile rubber suitable for oil and rated to a minimum of 200lbf/in^2 burst pressure and temperature rated to at least 150°C. Ensure that the

hose type and layout conform to *RAC Blue Book* specification.

The tank may be co-located with the engine or remotely (*eg* in the boot of a front-engined car). If the tank is next to the engine (*ie* within 4ft) the block-mounted breather scavenge unit may be removed and blanked off and the breather outlet pipe connected to the top of the dry-sump tank. Otherwise, retain the standard breather assembly and bleed to a catch tank. Try to keep oil hoses as level and direct as possible to reduce pressure loss.

The oil pressure relief valve should be set at 55–60lbf/in^2 @ 85°C. Where, for example, on the tank-pressure stage feed line the hose is ¾" bore and the pump union is only ⅝", use an appropriate adaptor as this creates only a local restriction. Scavenge stages on the return to the tank can be siamesed into ⅝" bore pipe.

Mounting and drives (13/29, 13/30)

The diagram (over) shows three alternative methods of belt drives for dry-sump systems. It is vital to ensure that the pump bracket is extremely strong (at least 6mm mild steel if fabricated) and accurately parallel to the crank axis, that the belt does not chafe and that it is protected from grit and stones, which will cause rapid wear to the pump pulley/belt teeth and even breakage.

A trapezoidal (rectangular) tooth belt is used with the Titan pump, ¾" wide. Automotive-pattern belts are stronger than industrial types. It is usual to allow for adjustment of the belt tension in the bracket design. Excessive tension will damage the pump front bearing and seal. Some care should go into the design of the pump bracket as sufficient space must be allowed for the size of the pump unions and access to the engine main oil feed gallery.

Blank pulleys can be purchased from gear manufacturers, which somewhat simplifies the design of the belt drive set-up. These can be machined with a lathe and appropriate flanges made to fit the water pump and alternator.

Oil, filtration and wear

Lubricating oil has to cope with contamination from a variety of sources and yet still maintain an oil film between the moving parts of the engine. Development of the exceptional oils available today is one of the great technological advances in automotive research in recent years. It is crucial to choose the finest quality for a tuned TC, both for running-in and for competition, and to protect the oil with a properly designed system so that it can perform its design function under optimum conditions.

The oil is subject to attack from solid particles inhaled by the engine if air filtration is inadequate, and from combustion products. Naturally, modern lubricants contain sophisticated additives to combat this latter effect, but if the operating conditions are not optimized, or the oil is neglected, the oil performance will be severely degraded.

Corrosion

Combustion creates water vapour containing oxides of carbon and sulphur, and oxides of nitrogen are also present. These acidic products can cause rust and consequent abrasion, and it is particularly important that the engine is not run too cold and that the warm-up phase is as short as possible (mildly tuned engines require coolant and oil thermostats) so as to limit the amount of condensation produced in the cylinder.

Coolant temperatures below 65°C greatly increase engine wear and reduce power *per se* by causing excessive oil drag. The GCT-recommended coolant temperature for all engines developing extra power is 75°C–80°C. These figures have been proven by extensive dyno-testing and strip-down/wear inspection, and combine satisfactory running conditions with a good factor of safety. With a race engine there is more to go wrong – and a lot less time to do something about it! *(see Case History No 10)*.

Corrosion can also result from oxidization of the oil. This is caused by neglect of change intervals and allowing the oil temperature to climb too high. That said, modern and especially synthetic oils have extraordinary resistance in this respect, Mobil 1 Rally being one outstanding example. When oxidization takes place, oxygen absorbed by the oil forms corrosive acids which attack bearings, which is the main reason why VP2 bearings were indium-coated.

Carbon and varnish deposition

Carbon forms naturally from the combustion process and deposits form on the piston crown, combustion chamber, valves (especially exhaust) and ring grooves. Formation of these deposits is limited by use of detergents in the oil (and latterly also the fuel). Dispersants in a good oil pick up contamination. Sludge forms when the dispersants break down. (The sludge found in TC cranks when the plugs are removed has been centrifuged out of solution and collected in the end of the drilling!)

If sludge is allowed to accumulate it can interfere with the filter – bearing in mind that most commercial types have a bypass valve, which allows unfiltered oil to circulate around the engine when the filter is blocked (or the oil is cold and running at high pressure); this sludge can circulate around the engine and cause abrasion damage. Formation is aggravated by running the engine oil too cold, *ie* condensation forms easily and is not evaporated off in the sump.

Varnish deposition is caused by oxidization of the oil and leads to fouling of rings and grooves and to loss of compression. Varnish, or lacquer as it is also known, also forms on the bore on high-mileage engines, which is one reason why cylinders must be honed and cleaned when re-ringing an engine.

Solid particles

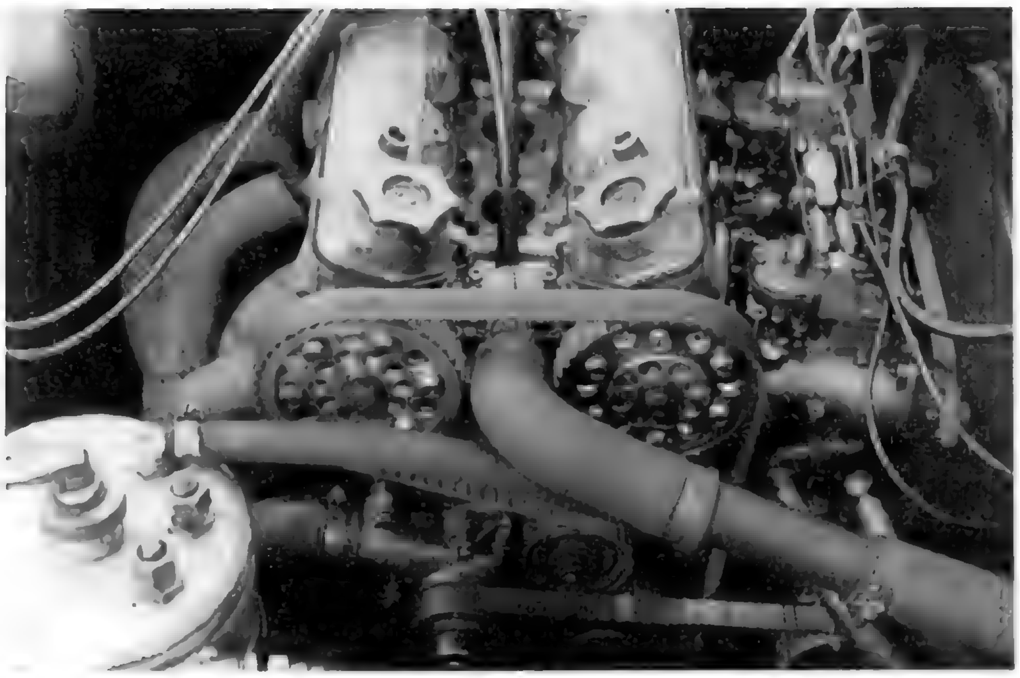
Solid particle contamination is primarily caused by inhalation of quartz dust if the air filtration system is inadequate, allowing foreign material to enter the engine during building or servicing, and broken metal burrs from engine components (especially during the running-in phase). The main particle size likely to cause most damage is in the 8–15 micron (1 micron = 0.001mm) range. Particles below 1 micron are too small to bridge the oil film and particles above 15 micron tend not to pass between the engine running components and exist in lower concentrations in air (except in extreme conditions). Quartz particles have a particularly abrasive effect within the cylinder because of the high combustion pressure. It is also worth noting that severe damage can be caused by excessive use of silicon gasket sealant during rebuilds – when it breaks loose it can clog the oil pump strainer.

Filtration

If filtration is inadequate, bearing and possibly crank damage is the primary result. Inadequate filtration can increase this wear rate by a factor of 5 or more and the problem of bearing damage is aggravated by the high bearing loads experienced on highly tuned engines (high CR, boosting and high volumetric efficiency all tend to raise the mep).

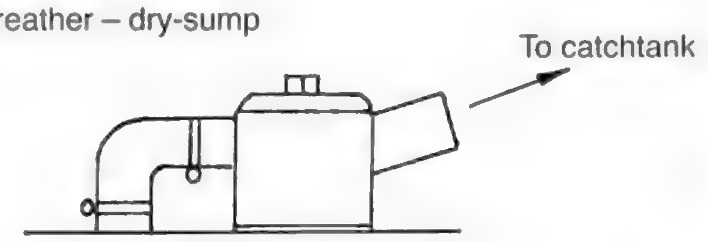
Oil selection

Viscosity is the measurement used to assess the oil's resistance to shear (*ie* disruption of its molecular structure) and hence its ability to prevent metal-to-metal contact. Failure of the lubrication system causes friction and overheating. The use of the Tufftriding process on early 2*l* cranks (introduced with the 132 2*l* and

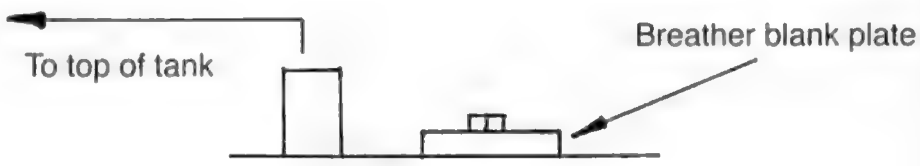


13/28: Dry-sump drive and oil tank of National Hot Rod driver Tom Casey's Peugeot 205. This is Tom's early engine (198bhp – see Case History No 3) which took him to very creditable 4th place in '94 World Championship and enabled him to retain '94 Irish title. Pipe on top (R) of tank goes to breather on block. Air filter is one-piece ITG.

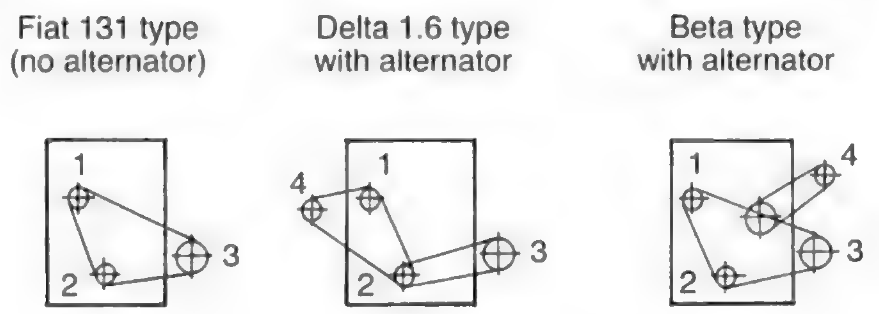
13/29: Block breather – dry-sump



Dry sump system: block breather set-up with tank remote from engine

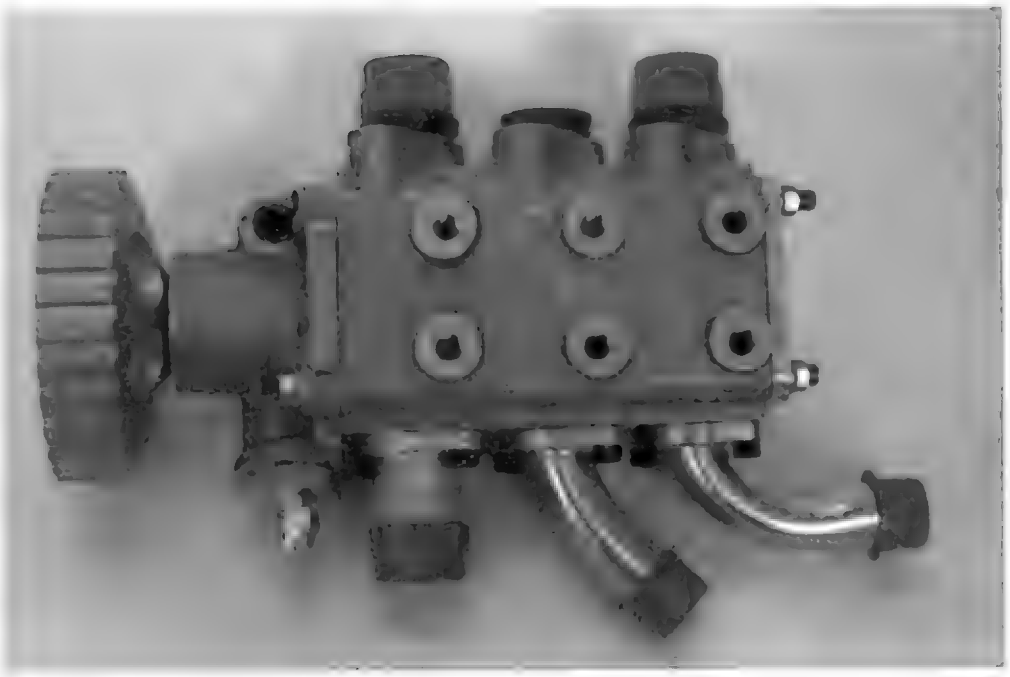


Dry sump system: set-up with tank located next to engine

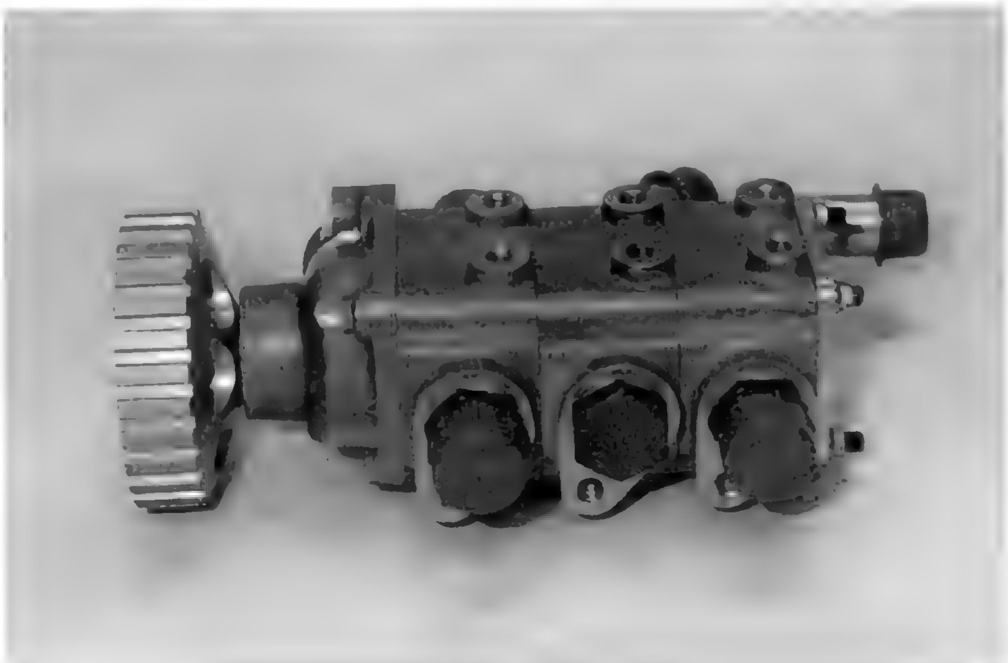


13/30: Alternative toothed-belt drive layouts for dry-sump systems

Key
1 Water pump (crank speed to 1.3 × crank speed)
2 Crank pulley
3 Oil pump (half crank speed)
4 Alternator (crank speed to 1.3 × crank speed)



13/31: Highly acclaimed Titan Series 2 rotor-type dry-sump pump; belt-driven at half crank speed, pump comprises 3 stages: front stage – pressure feed from tank to block (via filter); 2 rear stages are scavenge to return oil from sump to tank. Note adjustable pressure valve (lower left).

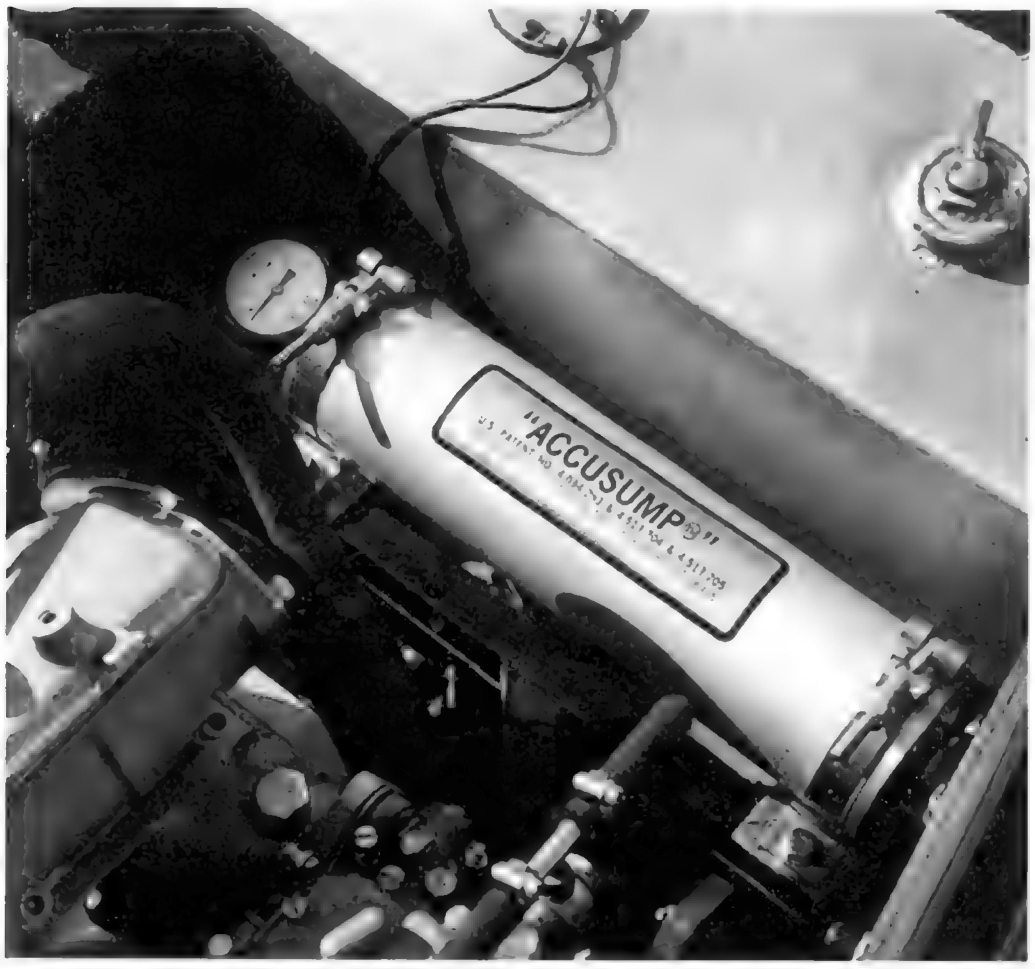


13/32: Top view of pump. Two rear outlets (scavenge) can be siamesed so that only one scavenge return line is used. Pump produces high volume with around 50–65lbf/in² pressure, 2.76 gall/1000rpm.

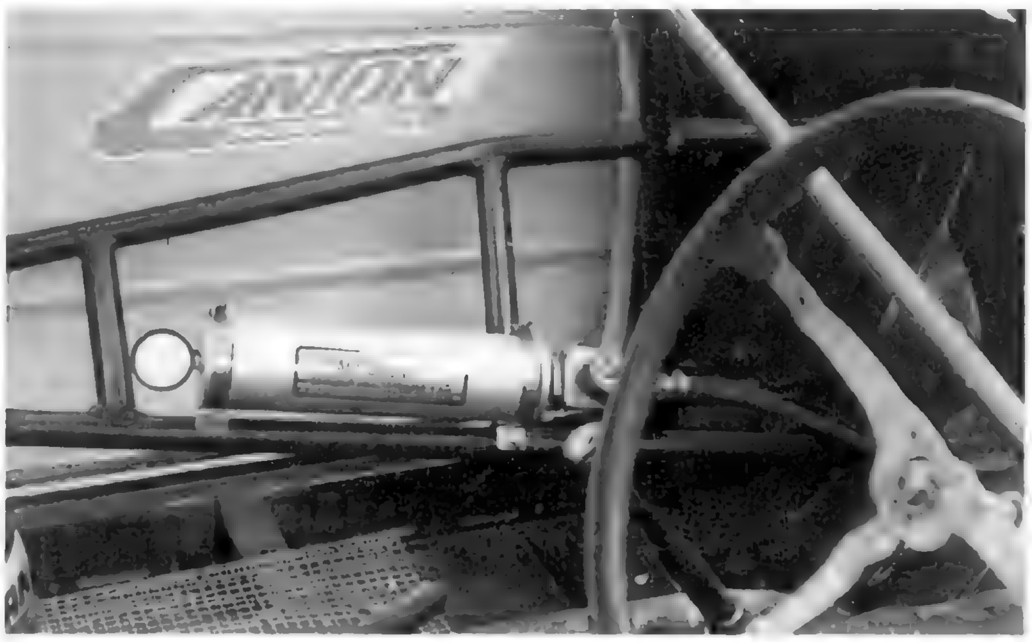
13/33: Accusump patented with pre-oiler valve by Canton/Mecca systems of USA. Alloy cylinder is roller-burnished – Teflon-coated; piston inside has air pressure at RH end, engine oil from lubrication system at other end. When engine oil pressure is high, piston compresses air pre-charge – if pressure drops, system feeds oil to engine automatically, preventing starvation. Surge – when pump goes dry and suddenly comes on load when oil returns to pickup, leading to excessive shaft load – is thereby eliminated. Model shown is 3qt type, approx 14" × 4¼".

13/34: 2qt electric Accusump with mounting clips; 12" × 4¼". Pre-oiling valve allows engine to lubricate prior to cranking – eliminating over 60% engine wear.

13/35: Turbo oiler: stores oil from feed line to turbo via special bleed valve. Releases oil to feed turbo bearings when engine is turned off. (Must be mounted vertically.)



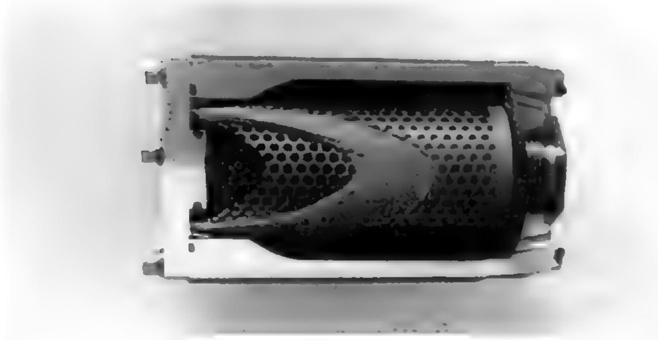
13/36: 1qt electric Accusump in Midtec Spider. Unit pre-oils engine (according to tests by American Society of Automotive Engineers over 60% of engine wear occurs during start-up) and provides excellent protection against oil starvation during hard cornering, essential on this high-powered road-race car.



13/37: 1qt electric Accusump installed in 1300 (standard engine) Toyota Starlet for Hot Rod. Despite its obvious advantages, Accusump is still regarded with suspicion by racers!

13/38: 1qt electric Accusump installed in Roy Carden's class-winning grasstrack car. Size of unit depends on extent of 'g' turns: short turns on oval/grasstrack mean 1qt is adequate for this St II 2l engine. Engine is switched off. Accusump has about 40lbf/in^2 oil pressure inside ready for next start-up. System easily has enough power to switch off oil pressure warning light!

13/39: Superior CM remote oil filter. Unlike commercial filters CM-type has no bypass valve allowing unfiltered oil to engine. Filter element, 4" high, is synthetic gauze, allowing deep filtration down to 8 or 4 micron depending on element type. Beautifully designed billet casing. Filter comes complete with mounting clip (as Turbo Oiler); connections are bottom-entry, side-exit ½NPT or ⅝NPT. Available as 15gall/min (shown) or 45gall/min. Working life of 15gpm model is 10,000 road miles or 750 race miles.

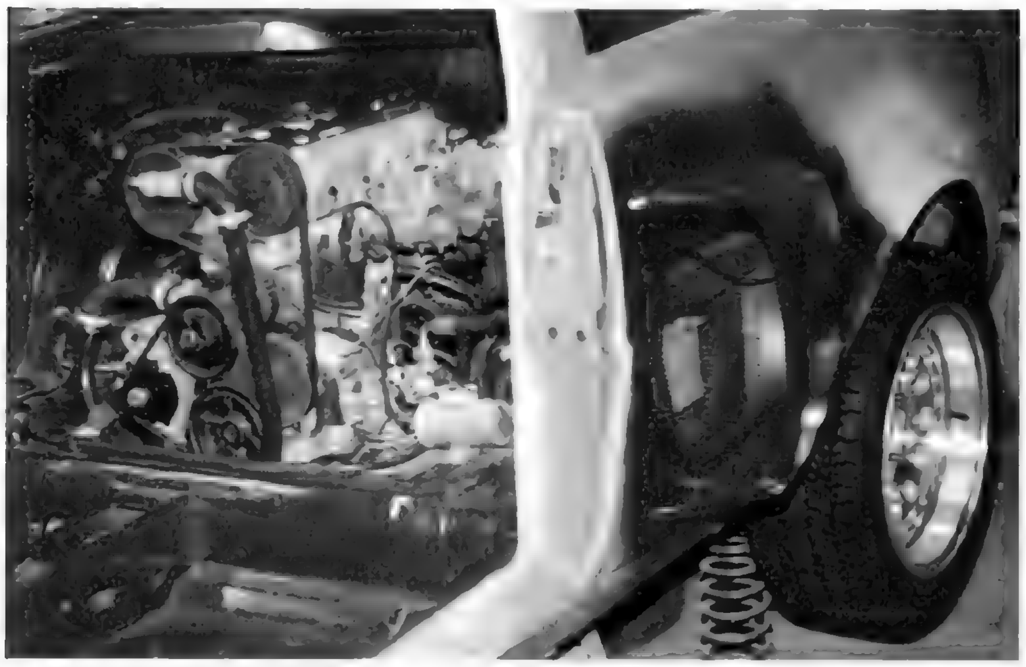


13/40: Cutaway of CM 45gpm oil filter (6" high) shows rigid perforated steel casing and woven synthetic element. Model is screw-on type. Note non-return valve between filter element and case – to stop oil siphoning back to sump when mounted horizontal or upside down.

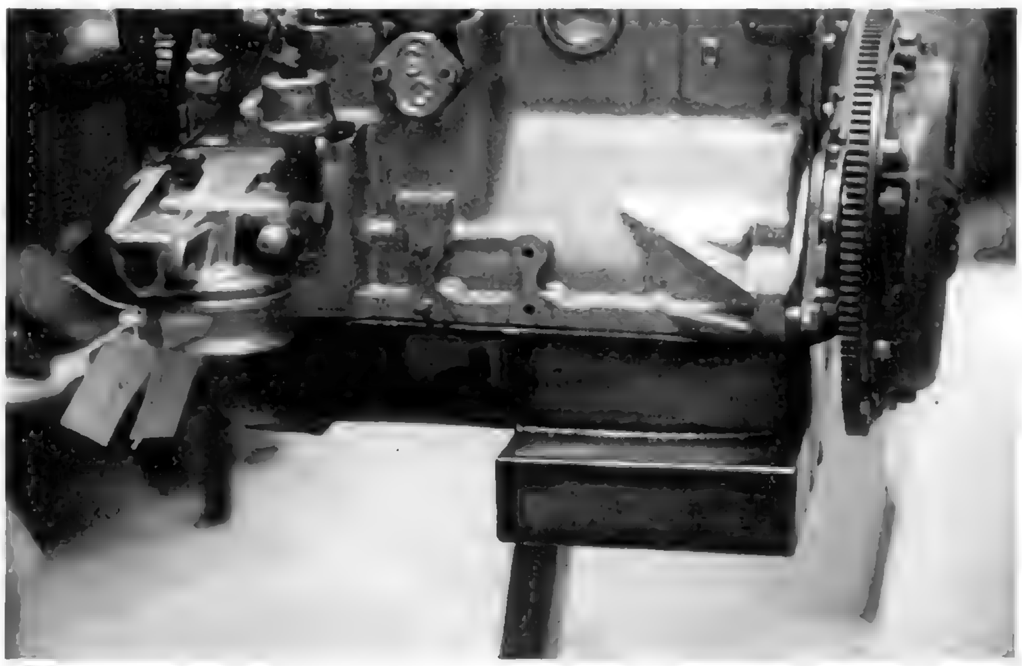
13/41: CM in-line oil filter, 6" long, copes with up to 45gpm and shares same features as spin-on/remote types.

13/42: Remote 45gpm CM oil filter with element. Whereas commercial paper oil filters can only filter to around 10–20 micron (μm), CM filters are available with 8 micron, or 4 micron, 'race only'. A clubman oil filter of this type has long been needed.

13/43: 45gpm CM spin-on filter with element. This filter can be used on production oil filter housing of TC, or Mocal remote filter head.



13/44: Large duct on side of Eric Cox's award-winning Skoda is for cold-air ducting to 15-row Serck oil cooler. Ducting is often neglected – air must be able to get in and out of cooler matrix.



13/45: GC built this sump for Ray Carden's Fiat 2l Toyota Starlet grasstrack car from a 105TC type. Intrusion of crossmember required use of Beta 2l pump, rear well and modified pickup. Worked well.

later changed to nitriding) improved their sliding/running characteristics considerably under these conditions, and it is quite usual to see such cranks in perfect condition at very high mileages (150,000 miles-plus), even though the aluminium-tin bearings may be damaged. This is due to the lower coefficient of friction induced by the process.

The SAE (Society of Automobile Engineers, USA) classification for viscosity uses the suffix 'W' to indicate that the viscosity of the oil (which varies with temperature) is given for –17.8°C (low-temperature viscosity) and 99°C (high-temperature viscosity). An oil with 5W/50 rating therefore has a viscosity of SAE 5W at –17.8°C and SAE 50 at 99°C. This designation defines an oil's ability to hold its viscosity at various temperatures and viscosity index improvers are commonly used in hydrocarbon oils to enhance their high-temperature performance and reduce low-temperature viscosity. Polymer molecules can be used to control the movement of hydrocarbon molecules as the temperature rises to reduce thinning of the oil.

Race oil must have good anti-foaming properties and anti-wear agents in addition to those qualities already mentioned, and it may well be said of oil (as with most things concerned with race engines!) that 'you get what you pay for'. The following short list is comprised of those lubricants 'tried and tested' at GCT and found to give flawless performance. Choose oils with classifications API-SH and CCMC-G5. These indicate the strength, detergent, dispersant and other major capabilities.

Running-in	*Racing*
Castrol GTX 2	Mobil 1 Rally
Valvoline Synthetic	Valvoline Synthetic
	Castrol RS

Oil temperature

Oil temperature and pressure are very closely related. Because of the narrow galleries through which the oil has to flow, the pump must meet a certain pressure criterion. When the oil becomes overly hot, it bleeds out from the bearings more quickly and the back-pressure in the galleries drops; this, of course, will show on the oil pressure gauge.

What is happening, in effect, is that due to back-leakage through the rotors and end faces, the pump cannot sustain an adequate flowrate to keep the pressure up inside the engine, and metal-to-metal contact is almost inevitable, even with an exceptionally good oil such as Mobil 1. The flowrate can be increased by modifying the relief valve, but the increased pressure that will result could blow off the oil filter when the engine is cold. Essentially, the pressure/flow output from all the pumps is more than adequate provided the oil temperature is kept sensible.

On GC dyno-tests, the rig computer is set to hold the oil temperature at around 85°C, and the Beta pump produces a steady 80lbf/in² at 7000rpm with this setting. But once, for experiment, the oil temperature was allowed to go up to 110° and the oil pressure dropped from 80 to 40lbf/in² – the engine would have run a bearing if the test had not been stopped.

There are two simple ways to reduce the oil temperature: one is to fit a heat exchanger in the oil circuit, which utilizes coolant from the engine to cool the oil (of course if the engine is too hot, it won't), and the other way is to fit an oil cooler, which relies on airflow *in and out* of the cooler matrix to cool the oil.

Either system can be fitted readily to the TC by means of a sandwich block fitted between the filter housing and the oil filter. These plates can incorporate an oil thermostat or an in-line type can be used. Keeping the oil temperature down is a very good insurance policy, especially when you remember that the seals on the TC are only temperature-rated to about 110°C–120°C.

Gauges

In order to monitor the lubrication system accurately, the following three instruments are essential:

1. Oil pressure gauge (preferably capillary type)
2. Low oil pressure warning switch, *eg* 25lbf/in²
3. Oil temperature gauge 0–120°C

Raceparts and Stack are two of the best. Without accurate instruments, it is impossible to diagnose a lubrication problem. The 124 Sport gauges, for example (used on Stratos replicas), were notoriously inaccurate, commonly showing only 4½bar when the true oil pressure was nearly 80lbf/in²! A low-pressure warning switch connected to a *prominent* red light on the dash is the only way to spot oil starvation – a gauge will not react quickly enough and in any case is impossible to read safely under race conditions!

Without knowledge of the oil temperature it is *impossible* to know whether low oil pressure is due to an inadequate system (pump, sump, etc) or excessive temperature, or to assess whether the oil is too cold, leading to power loss.

NOTES ON LUBRICATION CIRCUITS (WET SUMP)

1. Always use 'swept' fittings: cranked 90° types have a high flow loss (as much as 6ft of hose).
2. Hose type (see dry sump).
3. Remember that on the canister-type oil filter, the oil flows *into* the filter via the ring of holes concentric with the centre outlet.
4. Ensure that there is at least 5mm of clearance between the filter inlet holes and the underside of the remote filter head or flow will be severely restricted.
5. Oil thermostat should be fully open at approx 85°C.
6. Oil cooler selection:
   for 235mm matrix ½ bsp

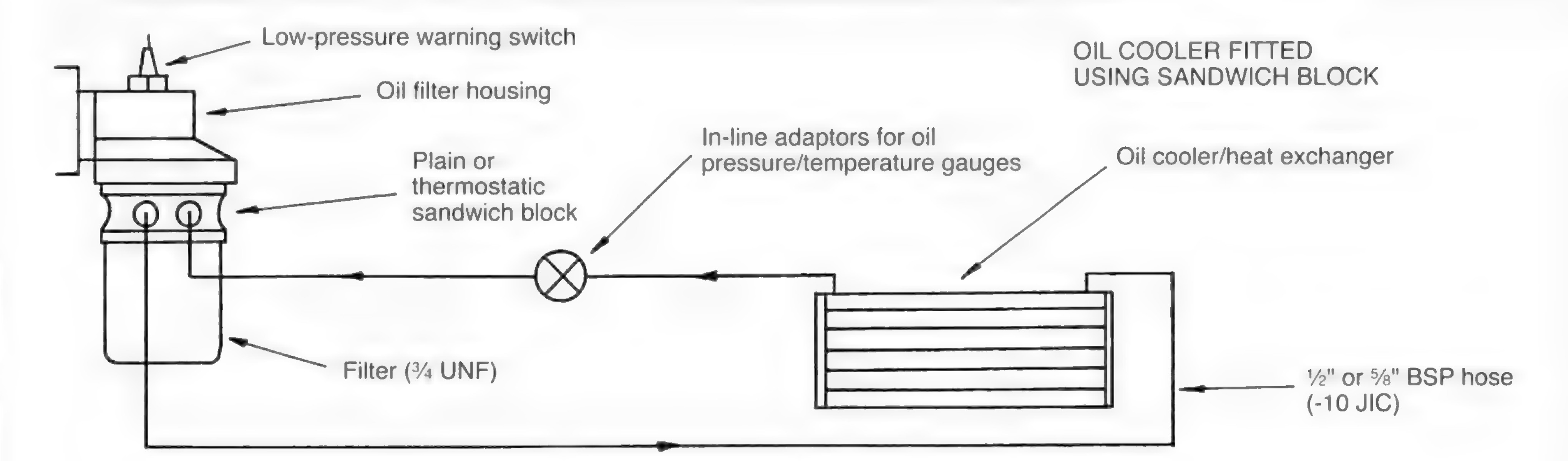
bhp	cooler size
100–125	10 row
125–160	13 row
160–180	16 row
180–210	19 row

   Above 210bhp, coolers can be used in multiples
7. Ensure that the cooler is located in an airstream to allow air in and out.
8. Low-pressure warning switch should be 20–25lbf/in². However, a well-used TC may well tick over with around 15lbf/in², so always use a gauge as well.
9. If possible, obtain an oil filter not fitted with a bypass valve – these allow unfiltered oil into the engine lubrication circuit.
10. Use a filter with a flap valve in the top to prevent oil syphoning back to the sump when the engine is turned off.

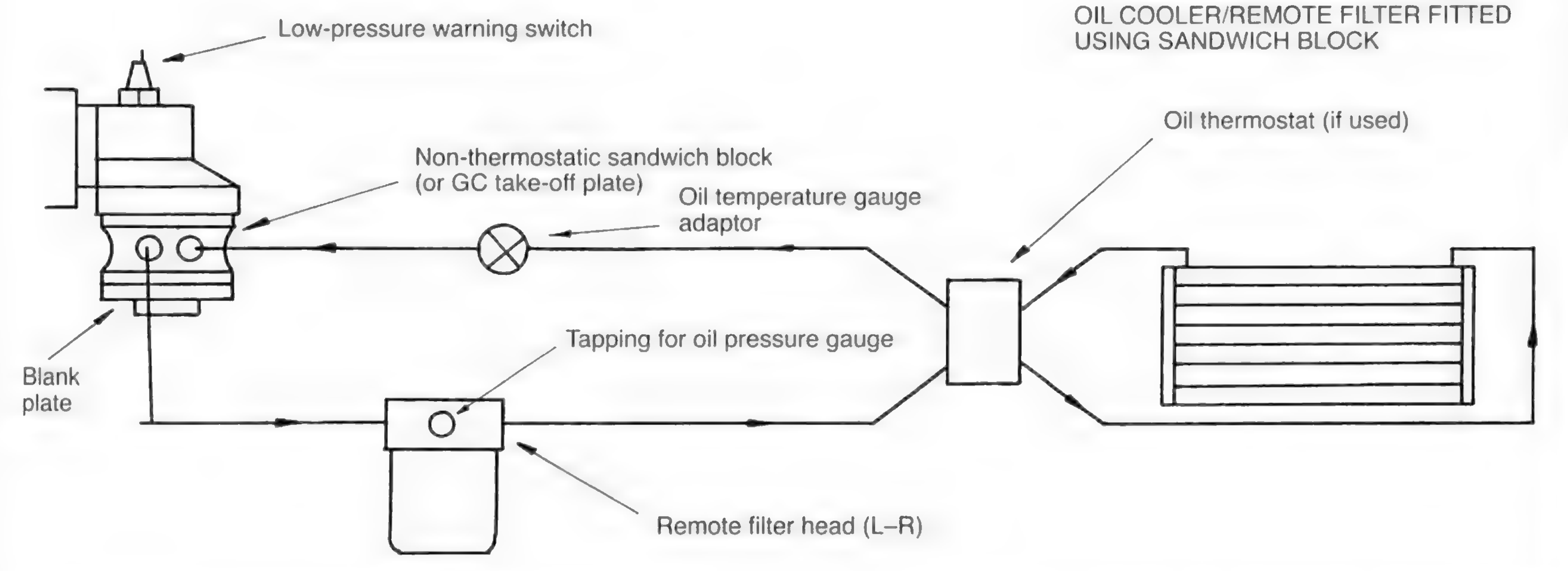
LUBRICATION PROBLEM AREAS

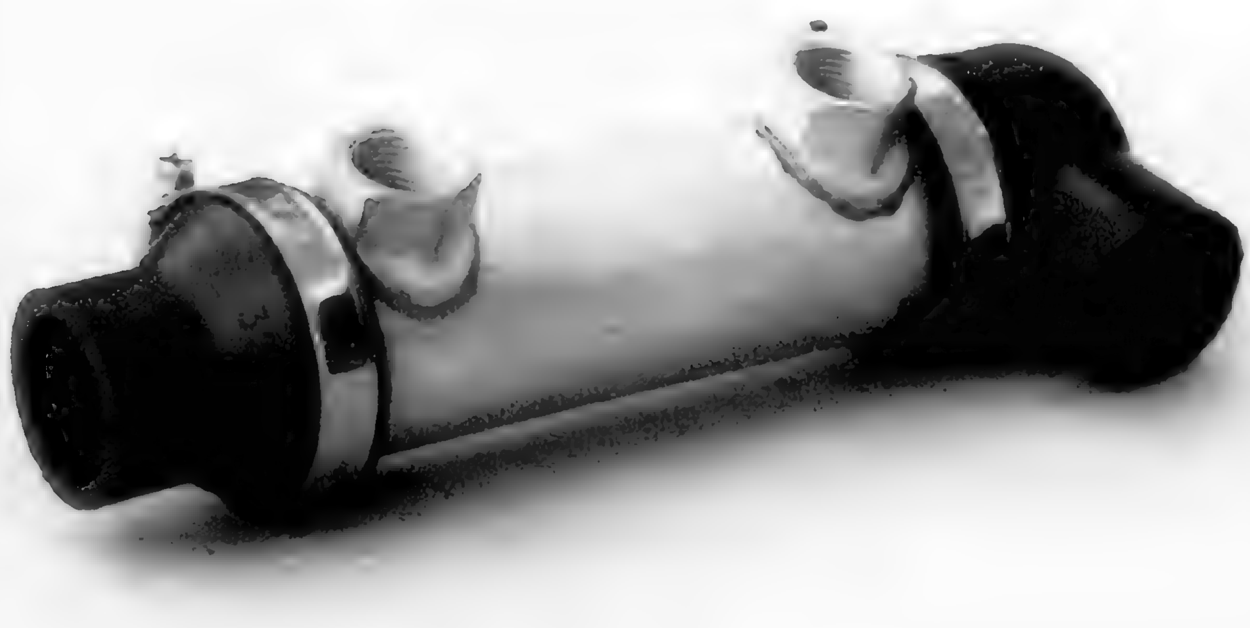
Fault-finding

Low oil pressure (hot):
- excessive temperature
- excessive aeration
- worn oil pump
- worn out splines on drive
- oil gallery plug dropped out
- broken auxiliary driveshaft or pump shaft
- low oil level in sump or tank
- inadequate baffling in sump
- blocked filter or lines connected wrongly
- worn or incorrectly set dry-sump pump relief spring
- belt drive worn or slipping
- oil hose collapsed
- faulty gauge
- oil lines too small
- damaged bearing or worn cam housing, crank/rod clearances too high
- sump base damaged, pickup starved
- filter too small (inadequate flowrate)

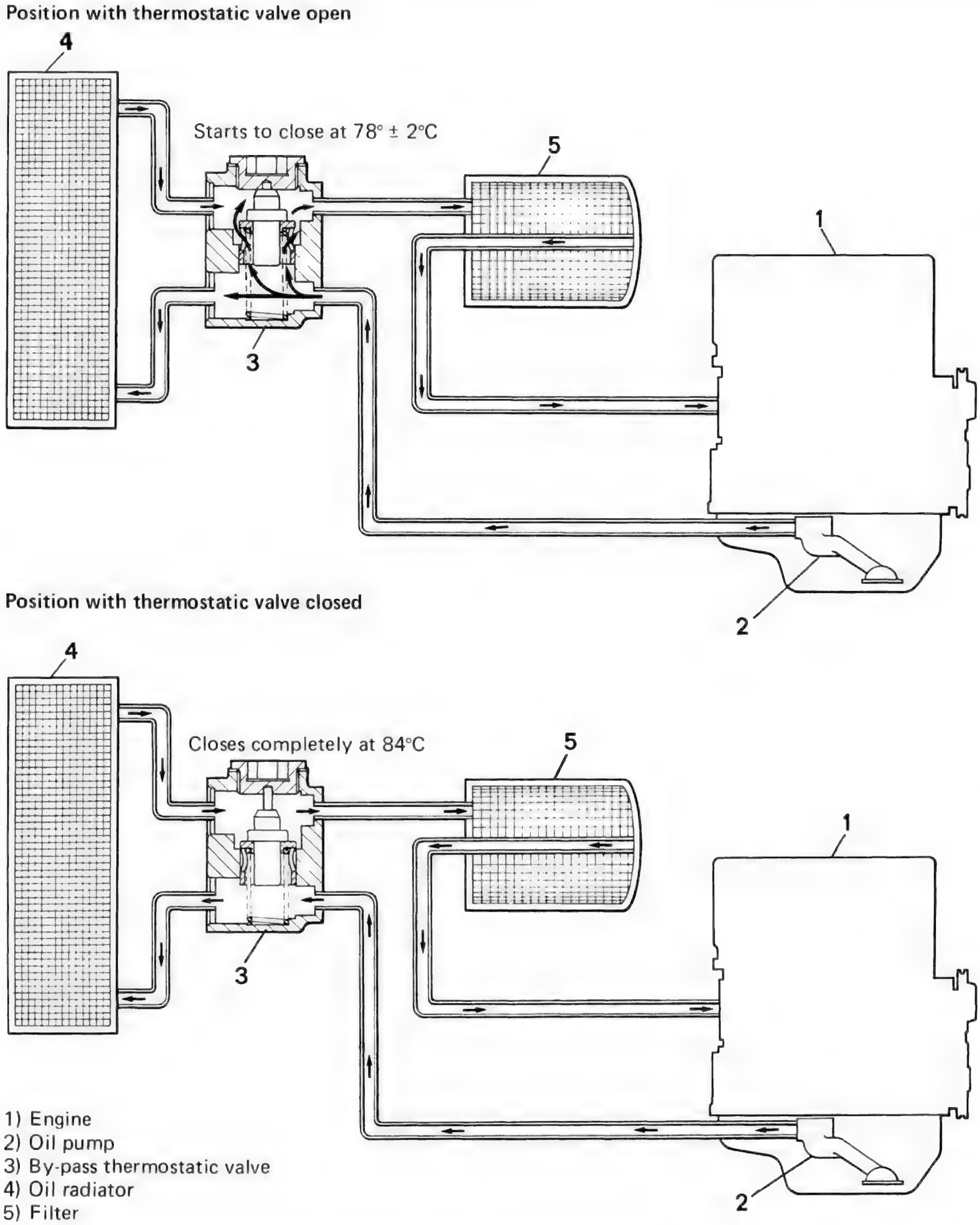


13/46: LUBRICATION CIRCUIT LAYOUTS



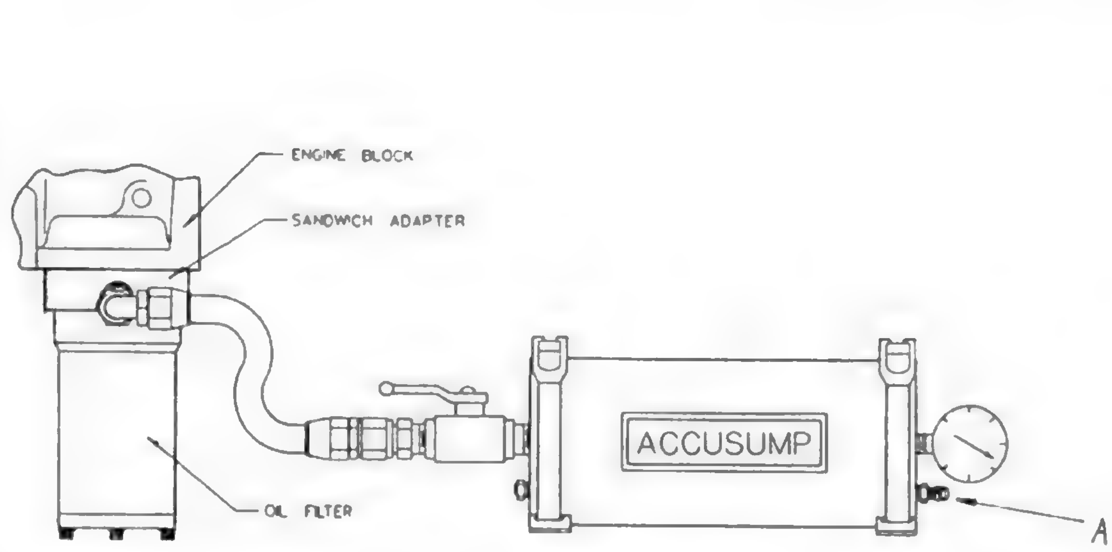


13/47: Coolant/oil heat exchanger made by Procomp Engineering of Birmingham is highly effective. Coolant flows along axis of unit, oil at 90°. Approx sizes are:
Standard (use up to approx 180bhp)
– 2½" dia × 9½" long
Large
– 2½" dia × 12½" long
Coolant ports are 1¼" dia, straight or 90°, oil ports ½ BSP female.
Heat exchanger is plumbed into return (cool) line from vehicle radiator; advantages over matrix (air) oil cooler are that oil reaches its operating temperature more quickly and unit is far less bulky and does not need to be exposed to airflow. This makes it an ideal choice for mid-engined vehicles like Stratos replica, Monte Carlo, where space is very limited and mounting cooler at front of vehicle would lead to high pressure loss in oil lines.

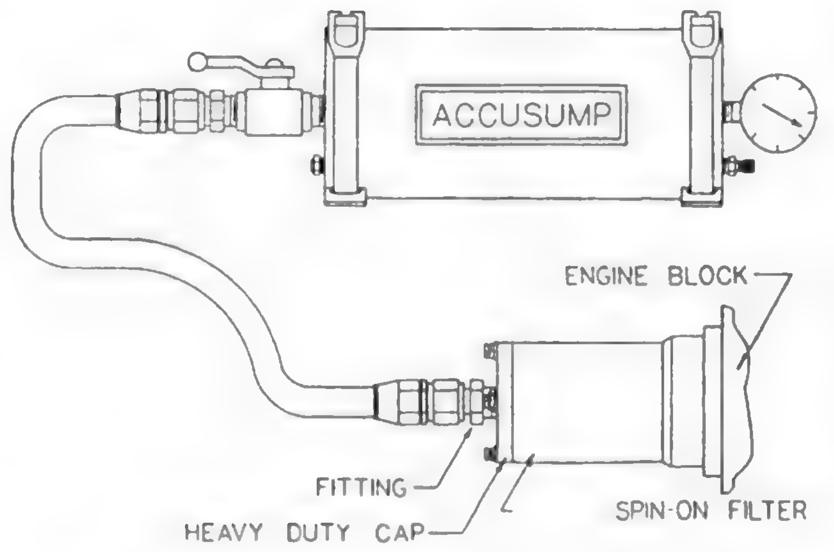


13/48: Diagram showing the engine oil cooling layout for an Abarth 130 TC.
(Fiat Auto SpA – copyright reserved)

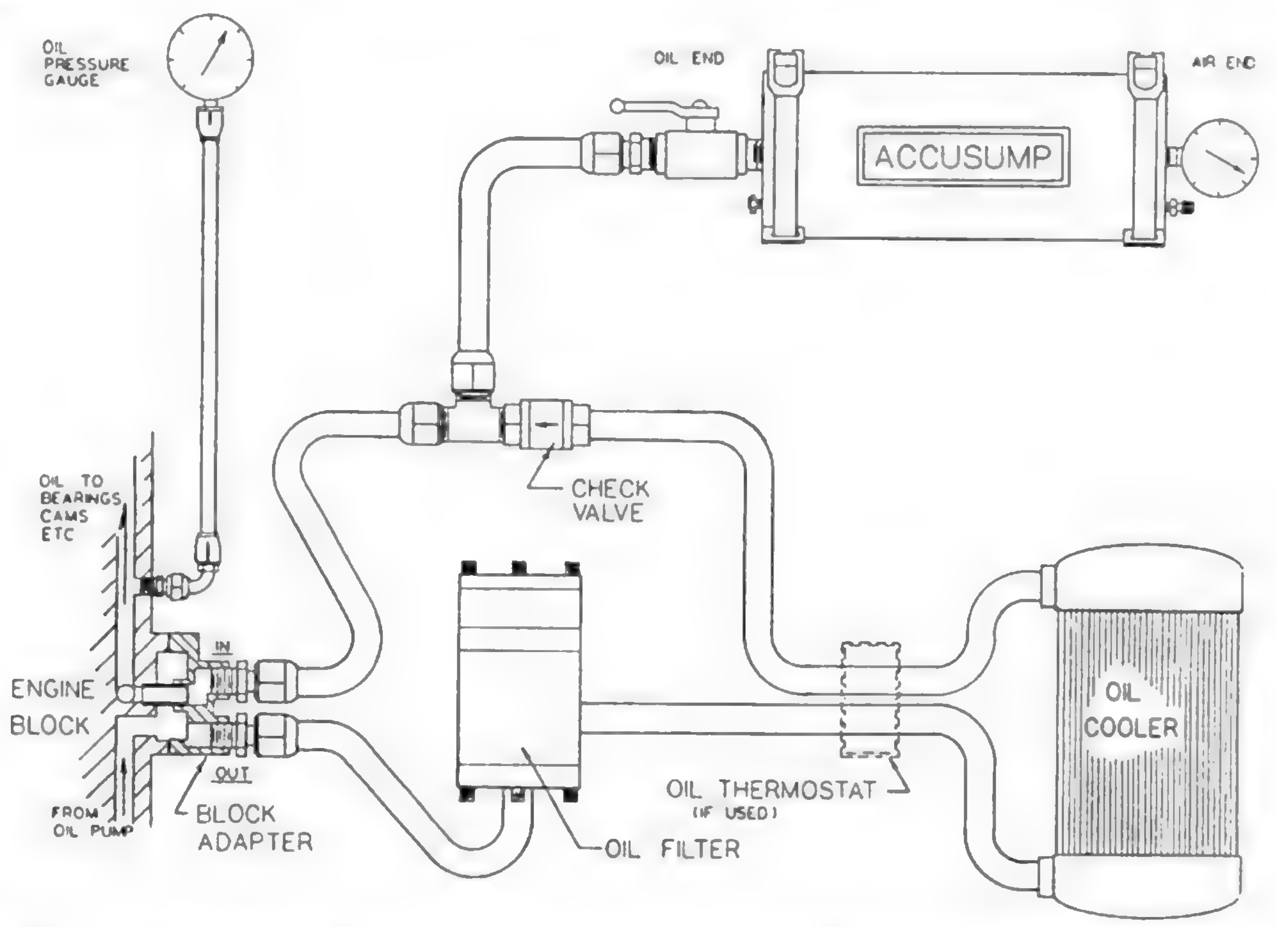
ACCUSUMP INSTALLATIONS



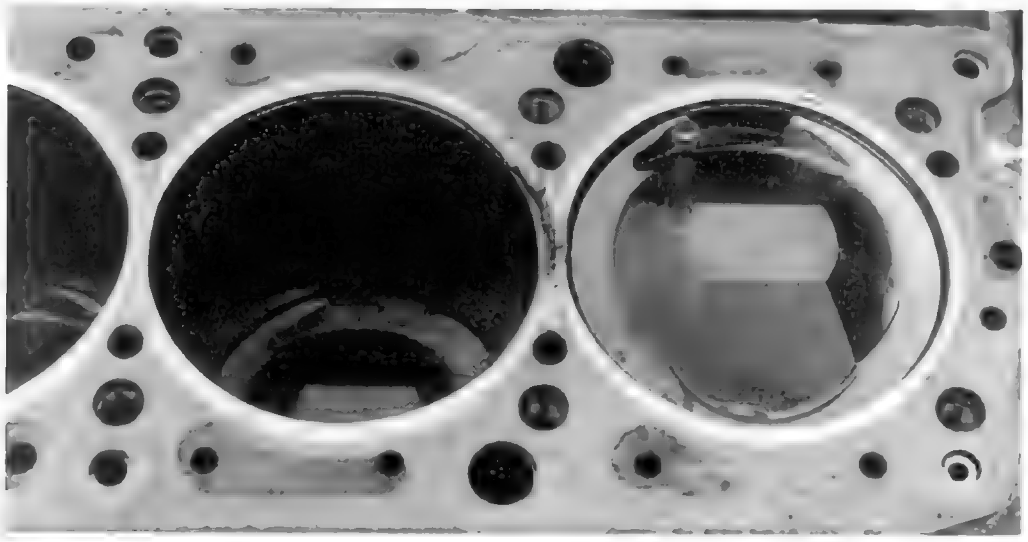
13/49: This diagram shows Accusump fitted with manual valve. Electric type is preferable since it can be operated remotely. It is advisable to shield valve 'A', which is an emergency blow, if unit is located inside car. Union to Accusump is ½" NPT (US National Pipe Taper).



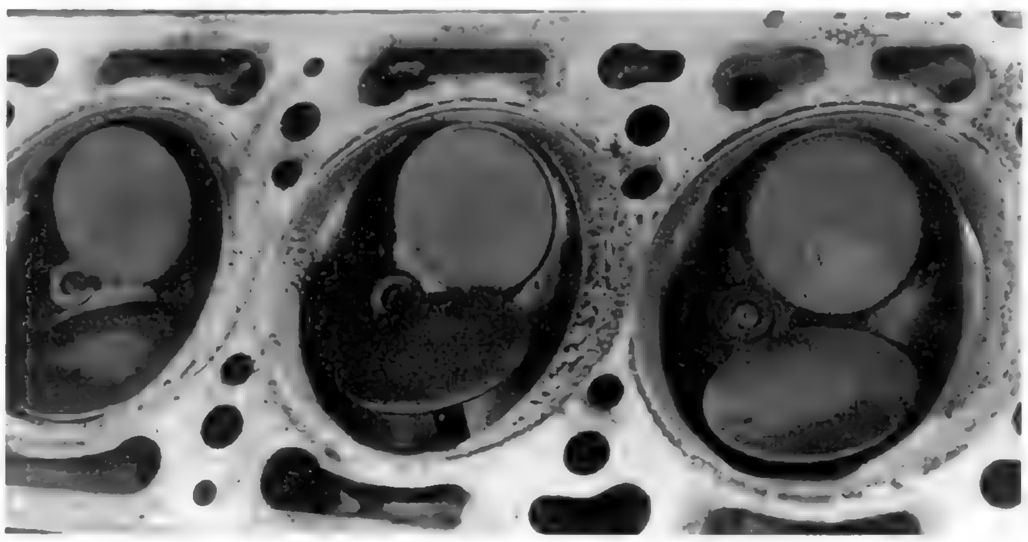
13/50: Canton/Mecca oil filters (spin-on type) can be fitted with special end cap as shown to allow system to plumb directly to filter.



13/51: Optimum layout above ensures Accusump receives filtered oil. Unit has very smooth bore which can easily be scored by impurities. Use of oil filter with a non-return valve makes use of check valve shown unnecessary.



13/52: "It's running too cold, I think I'll blank off part of the radiator", said owner of this Hot Rod 2l. Chronic detonation shows that perhaps he blanked it off a little too much.



13/53: Detonation damage to head looks worse than it was – head cleaned with 20thou" machining. More serious was problem of inadequate fit on inlet valve inserts which were tending to rotate! You can't race at this level unless your engine is 110% right.

– blocked (or broken) oil pickup
– dry-sump pump drive key broken/pulley loose
– pressure loss in oil system accessories

Oil leaks:
– gasket loose or not seated properly
– excessive crankcase pressure
– damaged seals or defective gasket
– hoses not connected properly
– breather system incorrectly connected

Excessive breather output:
– damaged bearing
– worn rings
– CR too high for breather capacity
– sump overfilled
– inadequate baffling in sump (windage)

Oil pressure too high:
– oil too cold
– blocked oil feed to head
– faulty relief valve (never seen at GCT)
– relief valve spring too strong
– blocked filter

Accusump installations
One of the simplest methods of connecting an Accusump to a TC is with a non-thermostatic sandwich block; one union is blanked off. The disadvantage is that the unit works on unfiltered oil with this layout. **(13/49–13/51)**

COOLING SYSTEM
The high torque produced by TCs and the use of an alloy head on the cast iron block mean that an adequate cooling system is vital to the survival of a tuned engine. Do not assume that the standard system (even with a radiator in perfect, uncorroded condition) will be up to the mark for more than about a 25–30% power increase. As with oil coolers, it is better to have a system with an overly large heat-dispersant capacity, where the radiator can be blanked off if required, than risk a system which is likely to cause overheating.

Running the engine at high temperature (100–110°C) reduces heat transfer from the combustion chamber and *theoretically* gives more power than a cool engine, and for this reason production engines are set up to run deliberately quite hot.

On tuned engines, a safety margin is vital to allow the driver to react if he observes the temperature exceeding a certain limit.

Raising the engine temperature raises the inlet air temperature (and the underbonnet temperature as a whole), reducing air density and hence power, and greatly increasing the risk of detonation. In summary, everything starts to go wrong!

In addition, high engine temperatures increase thermal stress on pistons and gaskets and greatly increase the risk of the head warping.

To operate effectively, the cooling system must cater for a number of criteria:

Pump
The pump must be of sufficient capacity to circulate the coolant at the required speed to match the heat generated by the engine. Hence, for example, normally aspirated engines have a lower cooling requirement at low speed than a Volumex (supercharged) engine. The cooling requirement increases with engine load/speed because the number of firing cycles per second increases. Therefore, a standard pump fitted to an engine producing standard peak power at 5800rpm may well not be adequate for a tuned version peaking at 7000rpm-plus.

Fortunately, GCT have found the high-capacity 2*l* 131 pump satisfactory on normally aspirated engines up to at least 230bhp; on turbocharged engines (*eg* 8v Integrale Gp A, approx 330bhp) the standard pump seems fine, although a well-sorted 1600 turbo would benefit from the larger 131 pump above 170bhp. The Volumex is already fitted with the high-capacity 2*l* 131 rotor.

V-belt drives are fine up to 7200rpm, but above this level a toothed-belt drive is essential as V-belts (plain or notched) are very prone to jumping off (usually straight into the cam belt!).

Coolant galleries/thermostats
The head gasket acts as a metering valve between the block and head to restrict the flow of coolant on the inlet side and increase it on the exhaust side, where the need for heat dispersion is greater. A fine balance has to be struck between the need to circulate coolant at high velocity by restricting the size of the engine coolant galleries and the danger of cooling fluid dwelling too long in one area and overheating.

Certainly there is benefit from enlarging the coolant galleries on highly tuned heads and radiusing their shape. This increases the surface area from which heat may be conducted to the coolant. Always ensure that the coolant galleries are clean: a weak acid solution, obtainable from radiator specialists, is useful for cleaning the block (take care not to spill it on the auxiliary driveshaft bearings).

GCT do not use high-temperature production thermostats on engines developing more than 30% extra power. The 131-type coolant outlet elbow will accept an 'in-head' thermostat (*eg* Unipart), which are available in a wide range of settings.

74°C is used on St II (165–175bhp 2*l*) for summer use, and 80°C in winter, when the ambient temperature is lower. GCT pure competition engines (St III-plus) do not use thermostats, but require a restrictor plate in the coolant outlet orifice in the head, which reduces the outlet size to around 20mm dia. The location of the outlet directly above the pump on early (non-reversed port layout) heads means that the coolant has a tendency to leave the head without circulating fully around the back of the engine. The restrictor plate helps to prevent this.

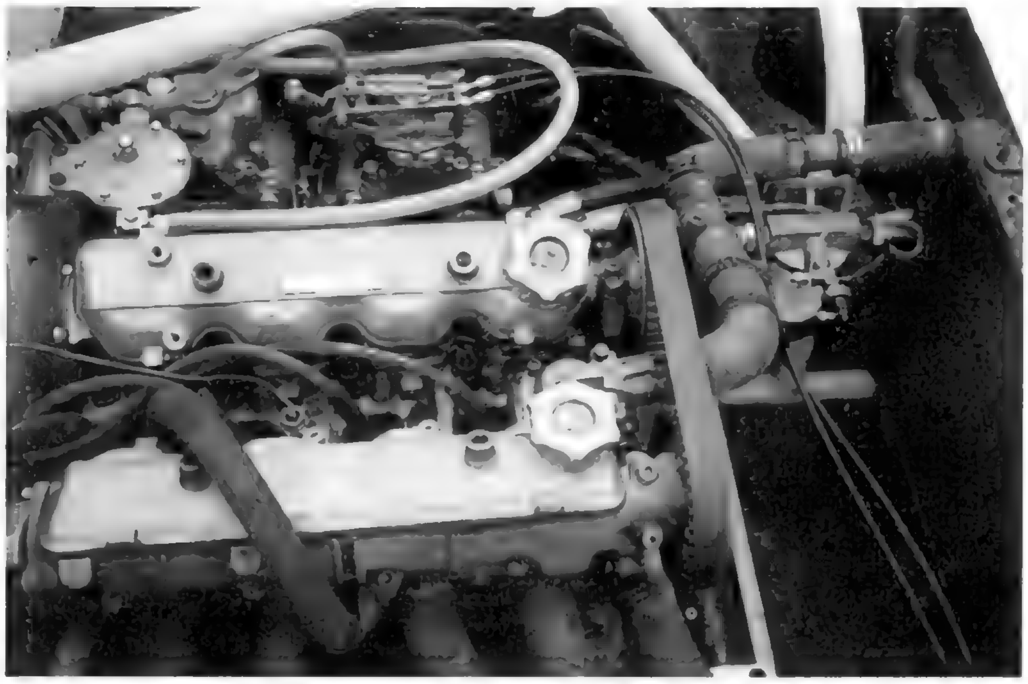
The production external thermostat is designed to circulate coolant around the head (but not the block) during the warm-up phase, so if an in-head thermostat is used it is important to drive the vehicle gently during warm-up to avoid overheating. In point of fact, in-head thermostats were used on early TCs. On turbocharged, mapped TCs one way to lower the temperature is to drill 4mm holes in the thermostat and road-test until the desired running temperature is achieved. If the coolant is run excessively cold, the fuelling may be upset.

Setting up engine temperature *vis a vis* thermostats and restrictor plates is very much a question of trial-and-error.

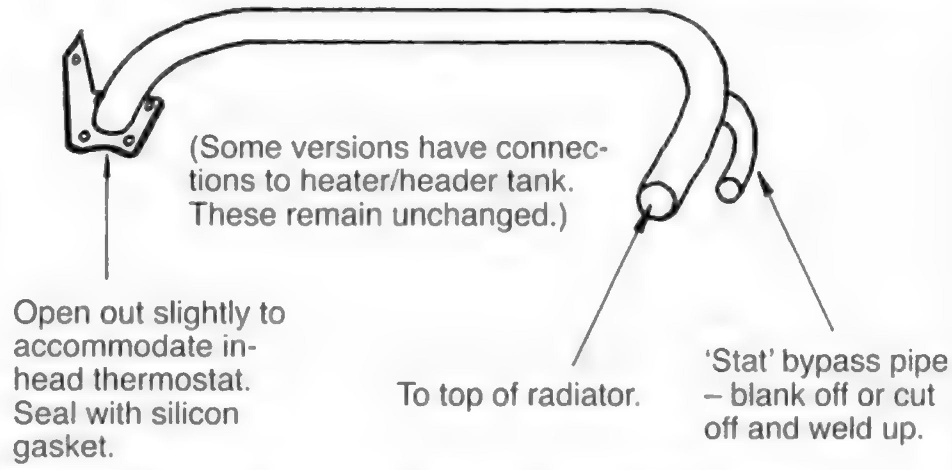
Radiator and hoses
Photographs throughout the book will give a good indication of the sizes of radiator required. As a general guide, if a production Fiat/Lancia rad is used, it will be large enough up to around 30% extra power; beyond that it should be modified to allow one or more extra cores. A high-capacity electric fan should be used to augment cooling at low speed (above 20mph it should cut out as the mass airflow over the rad alone should be sufficient). It is important to ensure that air passing through the radiator is ducted away from the engine.

The suction from the pump can cause hoses to collapse if they are too long. Race-quality silicon hose or Gates Vulcoflex, which is a superior wire-reinforced convoluted hose with formed ends, is ideal for short runs. Long runs of hose should be replaced with alloy or light-gauge steel tube. Always use high-quality clips – stainless steel types are the best. As with oil hose, coolant hose is subject to fatigue; this can break textile reinforcing, so whilst the hose may look satisfactory, it could be severely weakened. There are few things more annoying than losing a race engine due to a split hose!

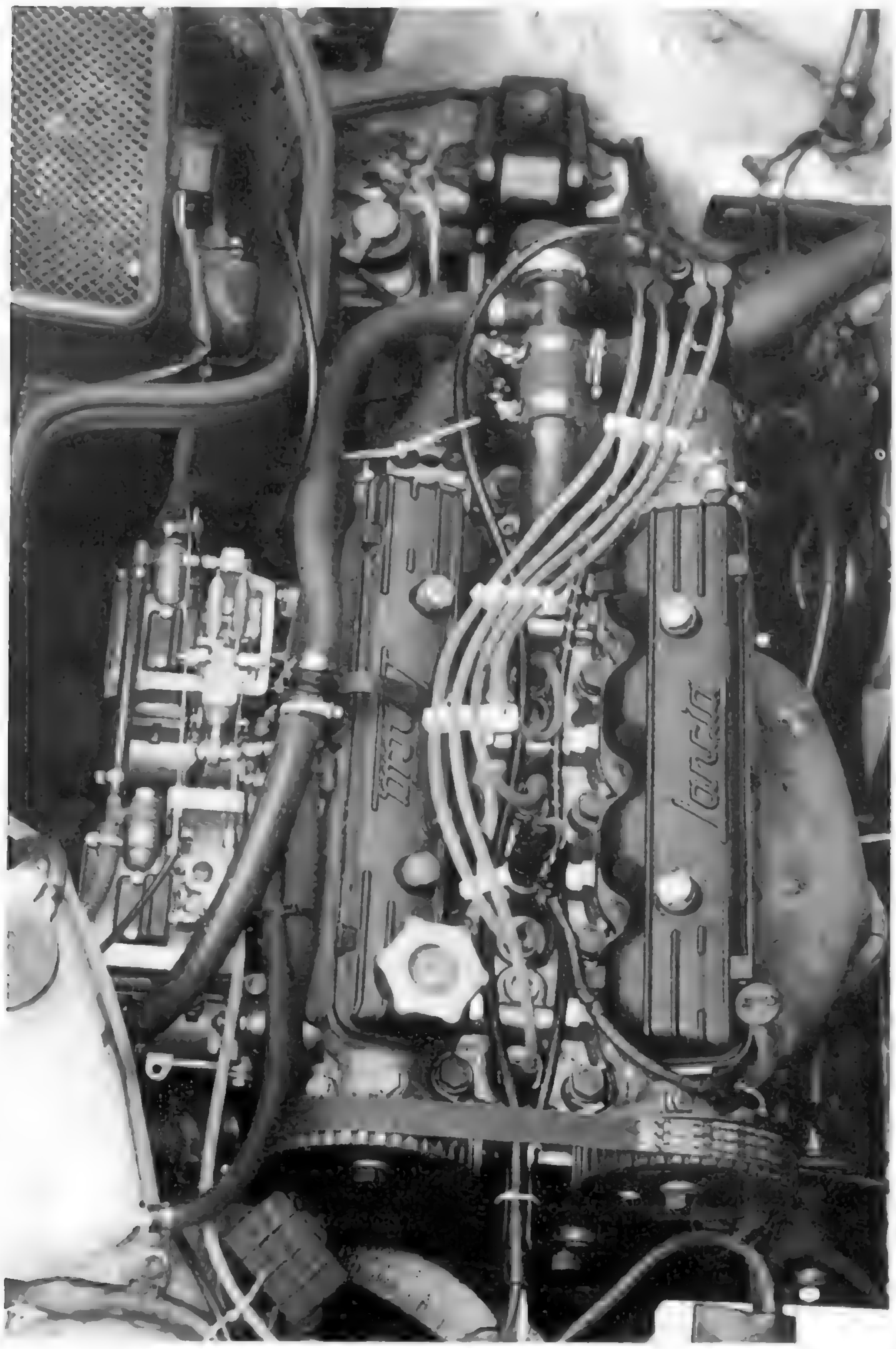
13/54: Early (fitted) and late 131 water pump pulleys. Late (dished) type is same as Beta. Pulleys must be used with correct (early or late) pump. Special short bolts may need to be used with pulley to avoid fouling pump body. If toothed belt drive is used, pump speed can be reduced by 30%. Always use notched, rather than plain V-belt if running standard set-up. Plain V-belts have nasty habit of jumping off at high revs!



13/55: Coolant outlet on 131 can be modified to accept top hose as shown on Tom Casey's Hot Rod. Note heater outlet at rear of head connected back to pump – helps keep rear of head cool. Remote filter head bolted to chassis carries T-piece for oil pressure gauge and low pressure warning switch. Twin oil filler caps were fitted, unusually on a dry-sumped engine, to allow cam boxes to be topped-up after car is trailered to races: 30° tilt on transporter causes oil to drain out – cams would start-up bone dry!



13/56: BETA-TYPE TOP RAIL IN-HEAD THERMOSTAT CONVERSION



13/57: Lancia Beta top rail modified to fit rear cooling outlet on Peter Gerrish's 2l Delta. Head is late (reversed-port) type: outlet was sensibly redesigned to help reduce overheating tendency around rear of head. End-drive distributor is useful on this type of layout and comes, in this case, from Uno, with modified curve.

13/58: Large duct forward of engine allows heat to disperse from radiator on this Fiesta Hot Rod. Radical spaceframe design by Autocross of Bracknell caused a sensation when first raced in 1994. (Adopted by Tom Casey for his Fiat engine in 1995.)

13/59: Twin-Cam X1/9 radiator layout was unsuccessful at cooling engine despite addition of two electric fans. Hot air could not escape from under-bonnet area. Louvres cured problem.

13/60: Keith Watson with his 1600 Fiat-powered Westfield. Build quality is obvious. Radiator would probably need to be bigger on tuned 2l version. Note electric fan for cooling engine at low (<20mph) speeds during traffic and to prevent build-up of heat in cooling system when engine is switched-off: thermoswitch in radiator controls function. When rad is lower than engine, as shown, a header tank must be used.

A pressurized cooling system should always be used, especially if the engine is raced at high altitude: as atmospheric pressure decreases, so does the boiling point of the coolant. The coolant must never be allowed to boil or heat transfer will be effectively nil. At sea-level, the boiling point of water in the cooling system increases linearly with the cap pressure setting; with a system pressure of 1.2bar (17.6lbf/in^2) the boiling point is raised to around 125°C compared with the boiling point of water at atmospheric pressure of 100°C. GCT normally use a 15lbf/in^2 system.

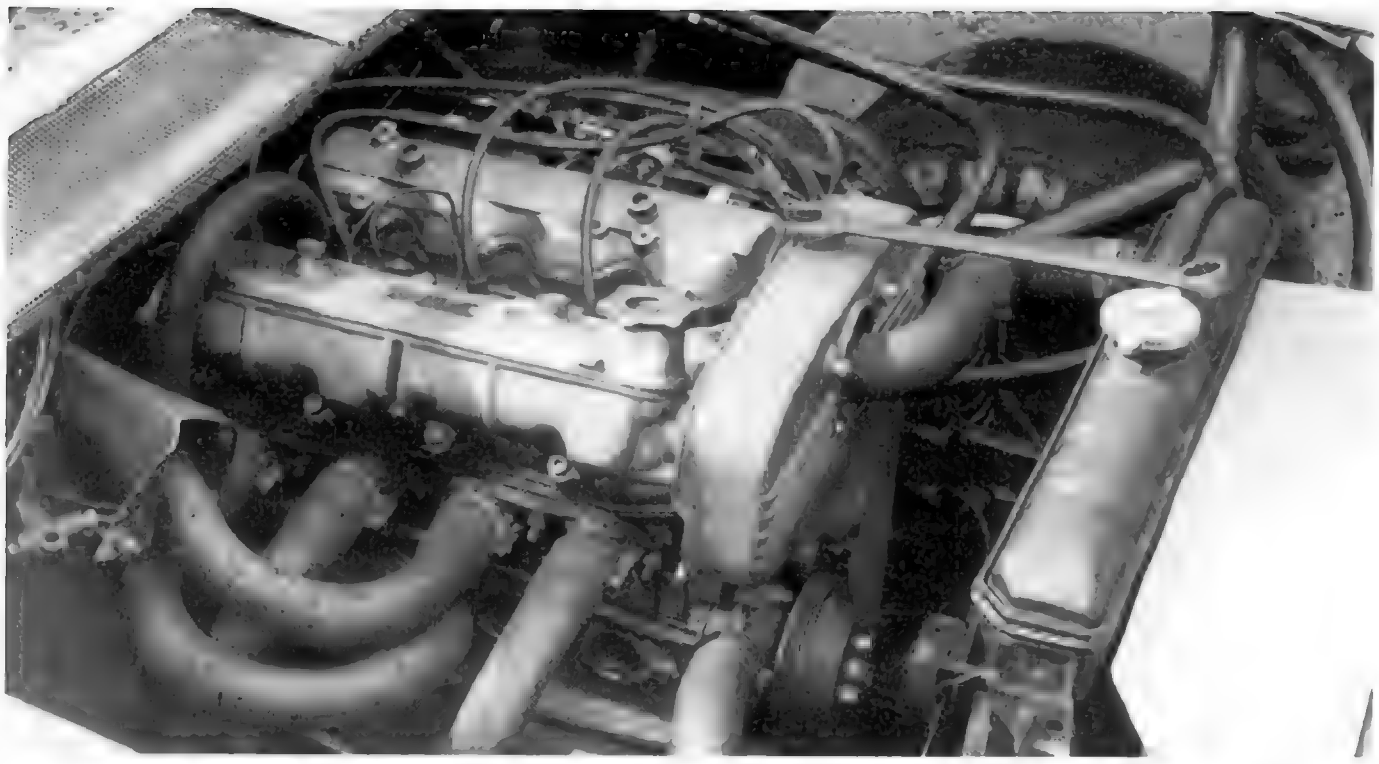
Boiling reduces the efficiency of the cooling system dramatically. The advantage of a radiator equipped with an expansion tank is that when the engine cools down, coolant vented into the expansion tank under hot conditions is drawn back into the system as the engine cools.

Another advantage of using an expansion tank is that the radiator can be mounted lower than the engine (since the system is filled at the expansion tank), otherwise the filler cap of the radiator must be at the highest point. Alternatively, a swirl pot may be incorporated into the top hose (from the head to the rad) which has a filler cap fitted. This allows vapour to be vented from the coolant, thus improving its efficiency.

13/61: Well sorted cooling system on Westfield Eleven built by John Hostler, Ted Cox and Dave Massey, features 2l water pump (on 1600 engine), Caterham Seven radiator, electric fan with thermostatic and manual override control. Short lengths of rubber coolant hose are complemented by aluminium sections on longer runs to prevent collapse. Radiator is fully ducted from nose aperture, air is vented to wheelarches. Header tank on top hose allows filling at highest point. Oil cooler is mounted behind radiator (13-row, 235mm matrix, ½bsp Serck), expansion tank is connected to heater rail.

Coolant

Antifreeze performs a number of functions. Essentially its job is to lower the freezing point of the coolant and raise its boiling point (so the production engine can be run hotter). However, it also disperses heat better than plain water because a correctly selected type (for alloy



13/62: Mk 2 Escort radiator provides ample cooling on Paul Thomas' National Hot Rod (see Case History No 8). Water pump is driven directly from crank via V-belt. Curiously shaped exhaust manifold is 4-1, could be better, but works OK when welds don't break. V-belt eventually jumped off and led to serious engine damage!

heads) scrubs off the stagnant coolant layer clinging to the head (and block) galleries. This prevents localized boiling. Fiat Paraflu is an ideal choice. The mix should be 30% minimum. Use of anti-freeze is also vital to prevent corrosion (which causes sludge build-up) of the head, block, pump and radiator. Anti-freeze also gives up its heat more readily than water due to its lower specific heat capacity.

Cooling system problem areas

Engine runs too hot:
- ignition timing wrong
- fuel mixture too lean (or defective carb choke assembly)
- plugs wrong heat range (too hot)
- fuel octane too low (causing detonation)
- intake air temperature too high
- pump too small
- belt slipping or drive ratio too low
- leaking circuit or level too low
- radiator too small
- thermostat too hot/restrictor plate too small
- hoses collapsing at high rpm (especially mid-engine layout)
- insufficient airflow through radiator
- cooling fan thermostat set too hot
- radiator position wrong (too high/low)
- radiator furred-up (or coolant galleries in engine blocked)
- blown head gasket

Coolant boils over:
- level in expansion tank too high
- blown head gasket
- pressure cap defective
- excessive underbonnet temperature (ducting required)
- fan not working
- thermostat defective

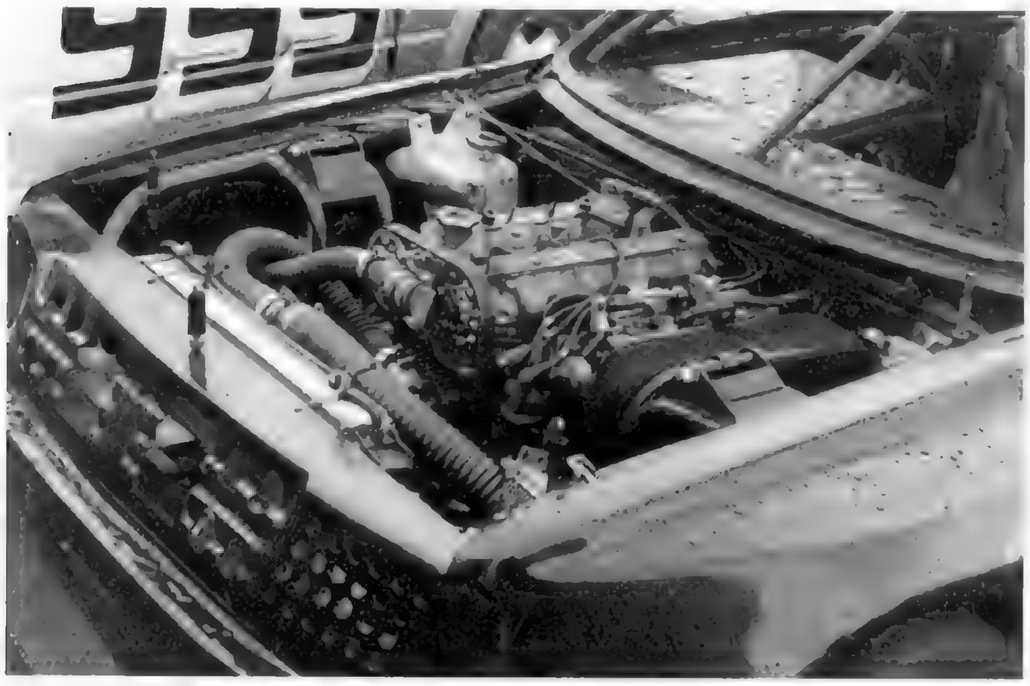
Engine runs too cold:
- thermostat too cold (or restrictor plate too big)
- radiator too large (blank off part of radiator)
- engine not producing maximum power

Engine freezes:
- antifreeze content too low

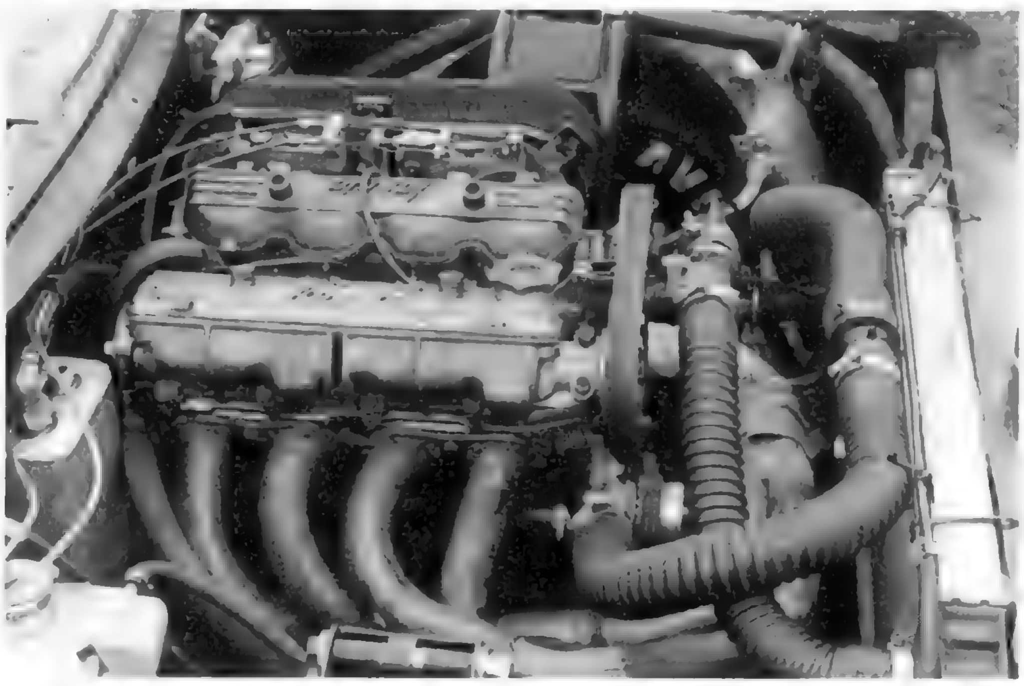
Influence of coolant/oil temperature on engine output

The engine featured in *Case History No 10* was tested with:
A Hot water (coolant) *ie* 82°C, oil 81°C
B Hot oil/hot water, *ie* 86°/82°C resp
C Hot oil – 86°C, water temp 71°C
The results for (corrected) torque, mechanical efficiency and bsfc are summarized in the table and plotted on the graph on the next page.

Important conclusions may be drawn from these results. Because of the heat transfer from the inlet port/manifold to the incoming mixture, the charge density is measurably reduced by running the engine at the higher temperature. Thermal coating would be an advantage, plus insulation between the manifold and head. It is vital, therefore, that the engine is maintained around the 73–75°C mark and inlet air temperature as near to ambient as possible. (These tests were run with a cell temperature of 13°C.) With hot oil (86°C) the pump load is significantly reduced (as indeed is engine friction) and the result (C) gives the best results overall.



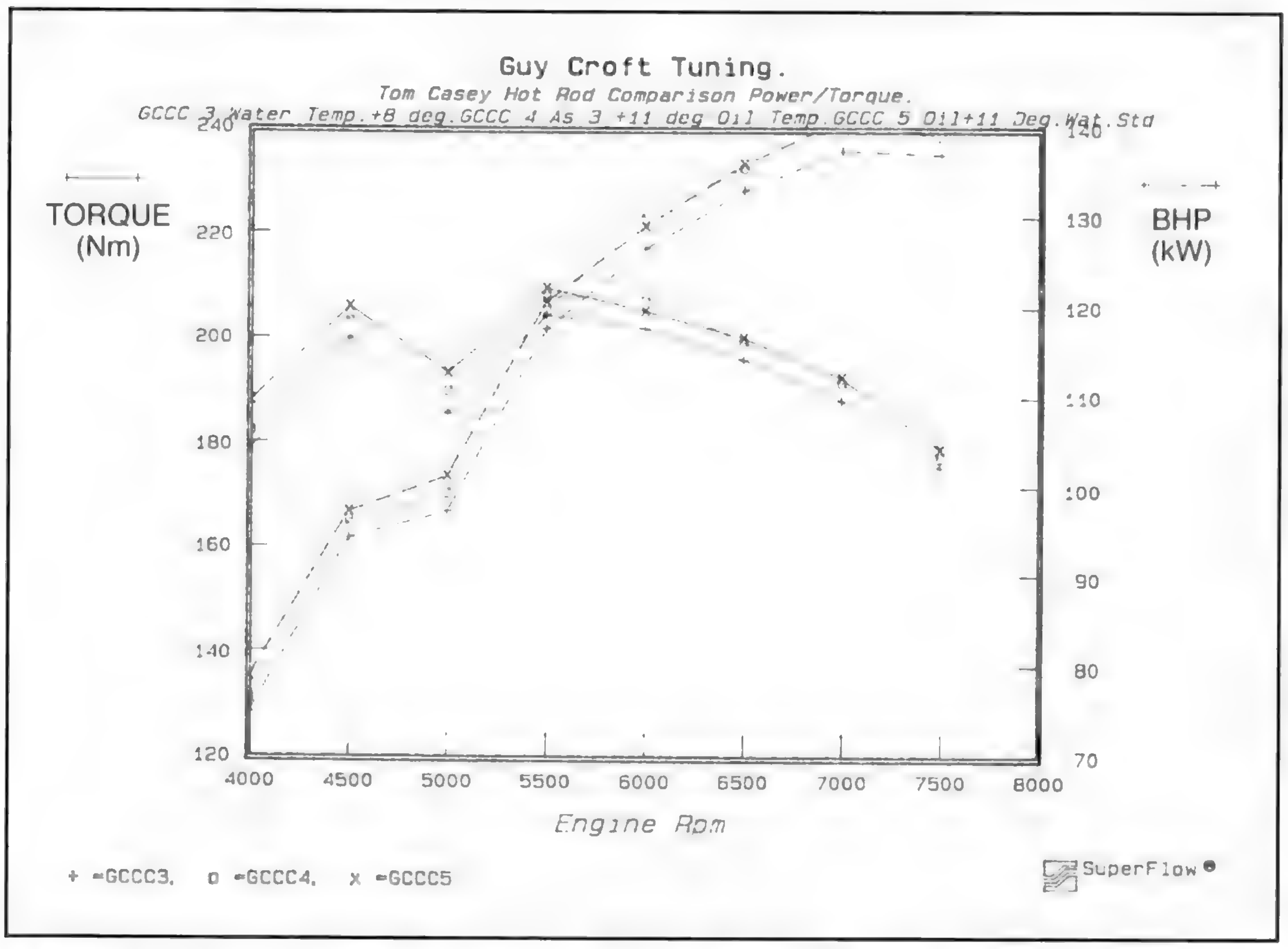
13/63: This Hot Rod has Porsche Turbo radiator...

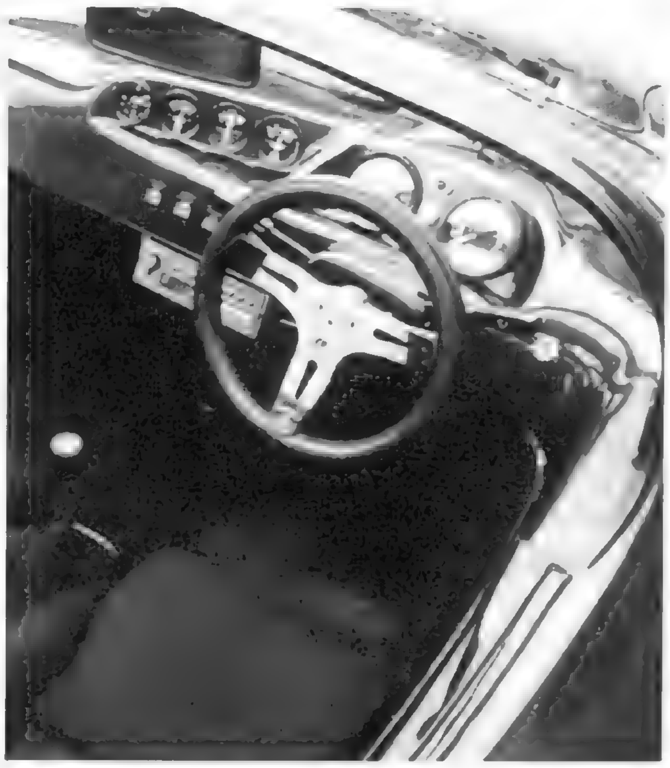


13/64: ...but vast hose array did nothing for system circulation. Rear coolant outlet on head (normally to heater) should be connected back to pump if heater is not used, to keep back of head cool. Layout shows space available for decent 4-1 manifold. Heat wrap on pipes can lead to early stress cracking.

Speed	Torque (corr) Nm			Mechanical efficiency (%)			Bsfc (gm/kW hr)		
(rpm)	A	B	C	A	B	C	A	B	C
4000	179.4	182.9	188.9	83.5	83.8	84.2	560	568	545
4500	200.7	204.5	206.4	83.7	83.9	84	456	455	418
5000	186.3	190.3	193.6	81.2	81.5	81.8	461	465	415
5500	205	208.6	**209.9**	81.2	81.5	81.6	375	425	375
6000	202.1	208.2	206.1	79.5	80	79.8	351	390	375
6500	195.9	199.5	200.5	77.3	77.6	77.7	441	403	391
7000	188.4	192.3	193.2	74.8	75.2	75.2	502	358	410
7500	175.3	177.5	179	71.4	71.7	71.8	469	356	356
(209.9Nm = 155.3lbf ft)									

The equation: Power (kW) = $2\pi n$ (radians/sec) × Torque (Nm) leads to a power output at 5500rpm of 120.9kW, *ie* a bmep of 186.4lbf/in^2, the highest achieved by GCT at time of writing. Note that, generally, test C also gives the best results for mechanical efficiency.





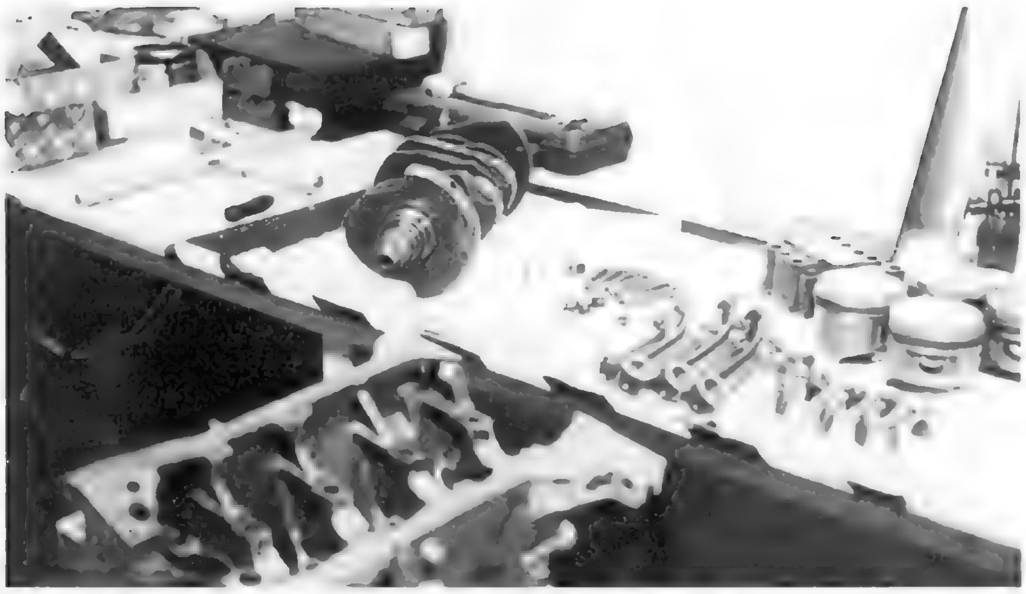
13/65: Proper instrumentation in Midtec demo car. Oil temperature, pressure, coolant temperature and low-pressure warning lights.

13/66: Tidy engine bay of Roger Smith's FSO-Fiat TC. Unit above rear carb is Kenlowe cooling system heater – as Roger says "I plug it in first thing in the morning, go and have a cup of tea, by the time I've finished, the engine is warmed up!" Standard FSO radiator copes perfectly with 145bhp-plus of 2l engine. Pipercross air filter socks fitted over rampipes are good for about 155bhp. Ignition pack is Marelli electronic – amplifier unit is mounted on rear of finned heat sink to right of expansion tank. If you encounter curious electrical misfire around 70,000 miles, amplifier unit is probably cause. Replacement items by Unipart are a lot cheaper than OE items.

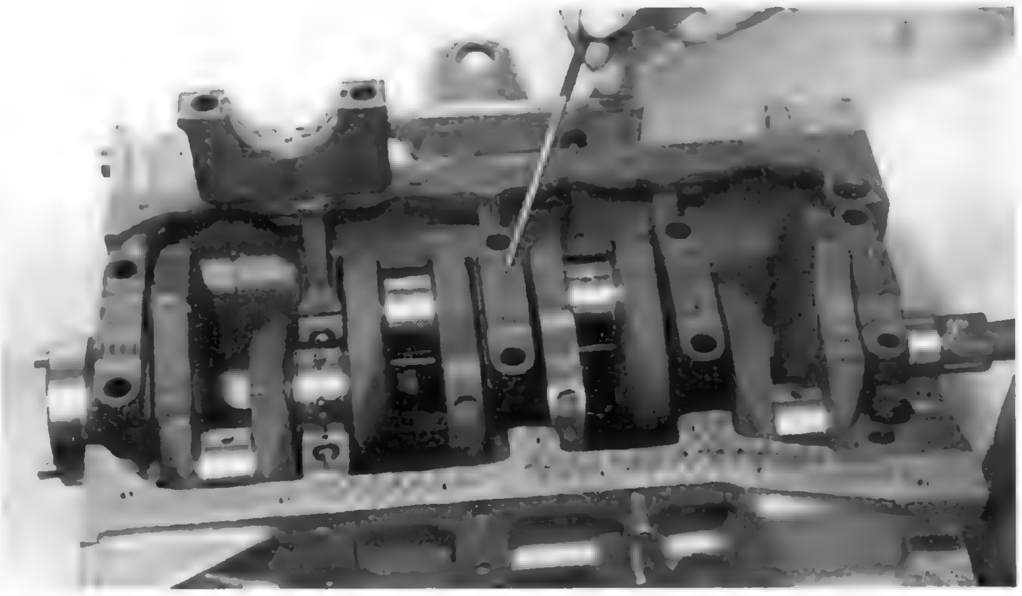
CHAPTER 14

BUILDING UP THE ENGINE

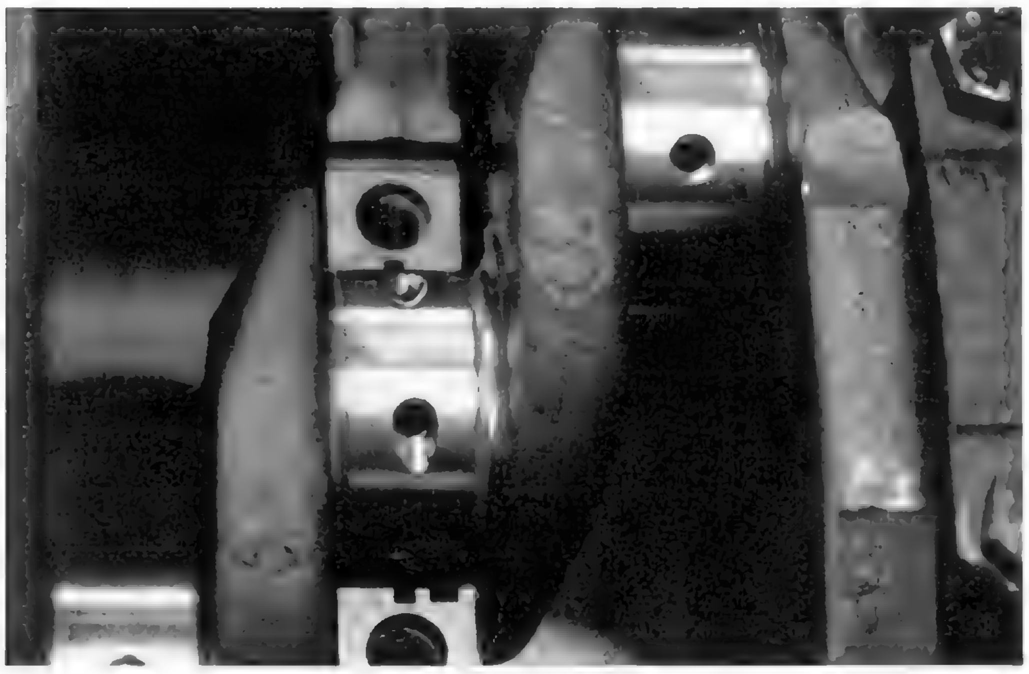
Including dry-building



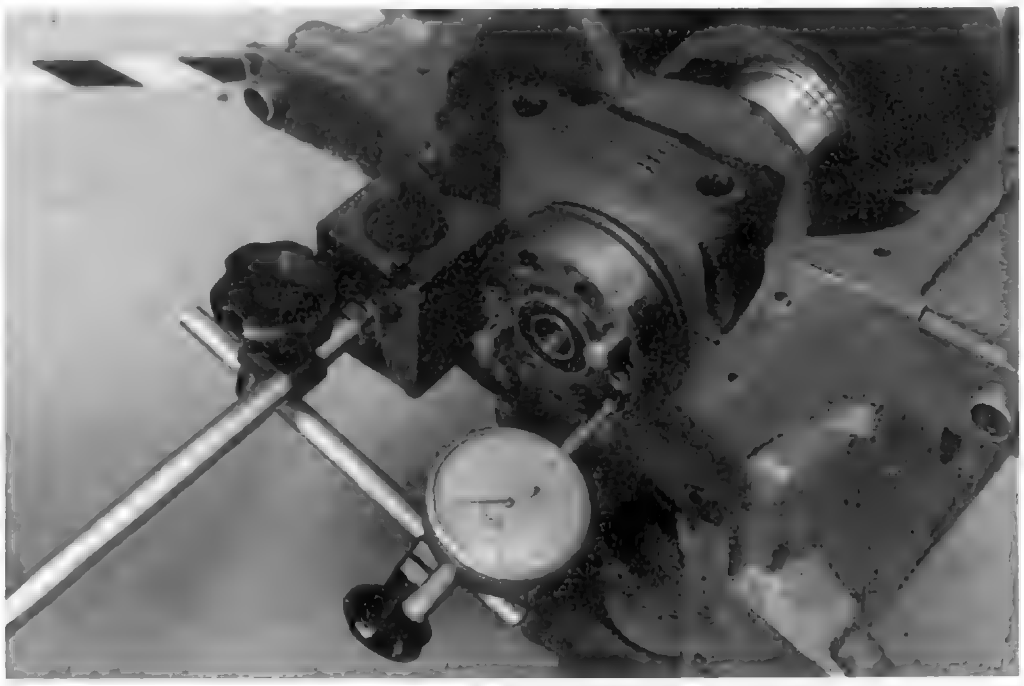
14/1: Before starting, lay out all parts on a clean bench. Build-up with pistons can easily be achieved in under an hour in these conditions.



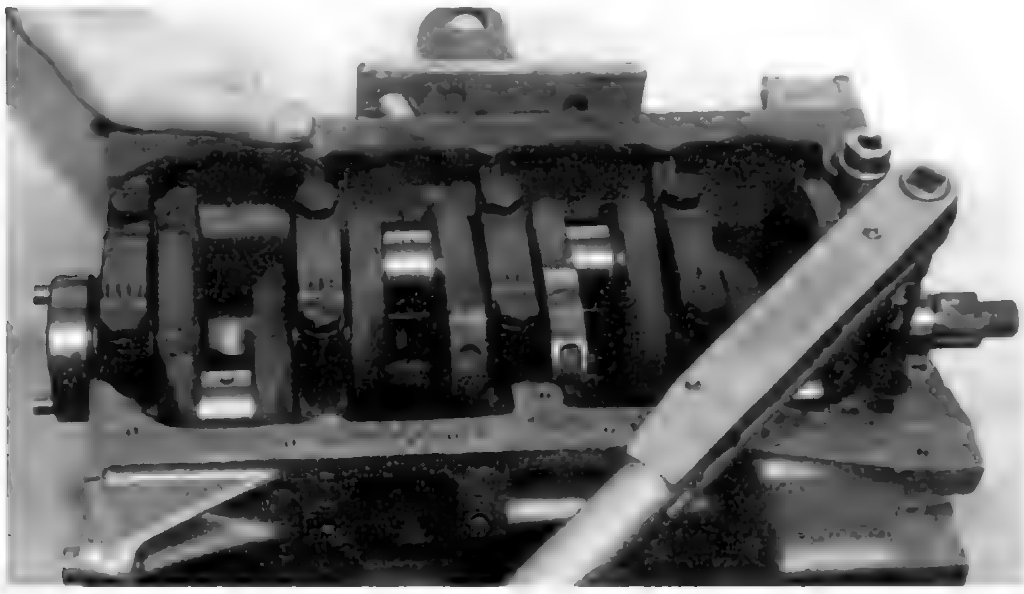
14/2: Thrust washers are installed with oil groove facing crank. Lubricate shells with graphite grease, oil crank journals and lay crank carefully in place. Centre main cap carries wide bearing. Caps are numbered to block – don't interchange because they are line-bored at factory. No 1 cap (at right) is not 'notched'. As seen in this shot, No 2 cap has one notch, centre cap has 2 etc. Note scrupulous cleanliness of whole assembly. Engine must be built in a clean, dust-free area.



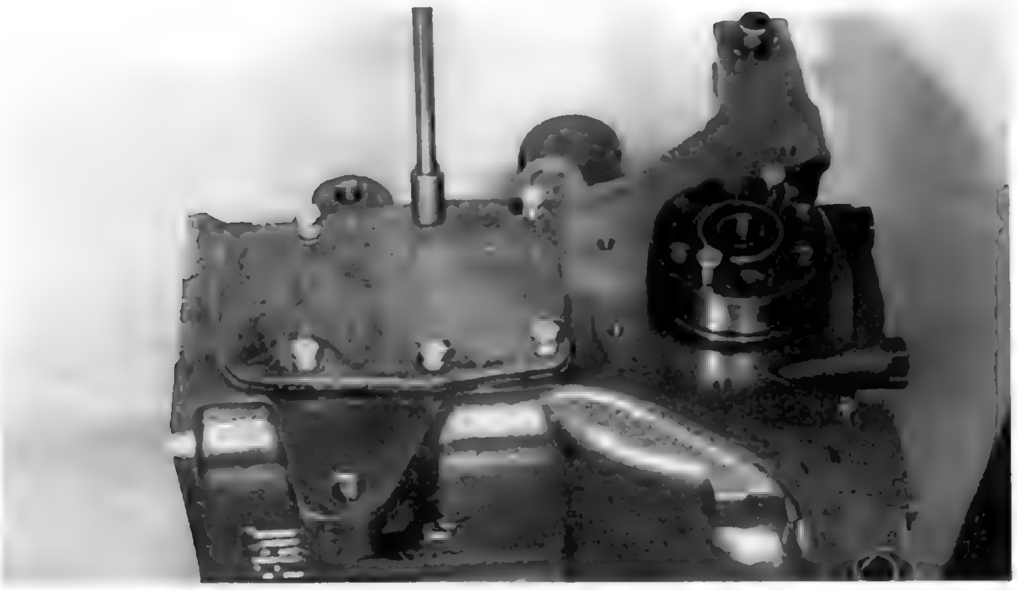
14/3: Radius on crank big-end and keyholing detail on crank main journal.



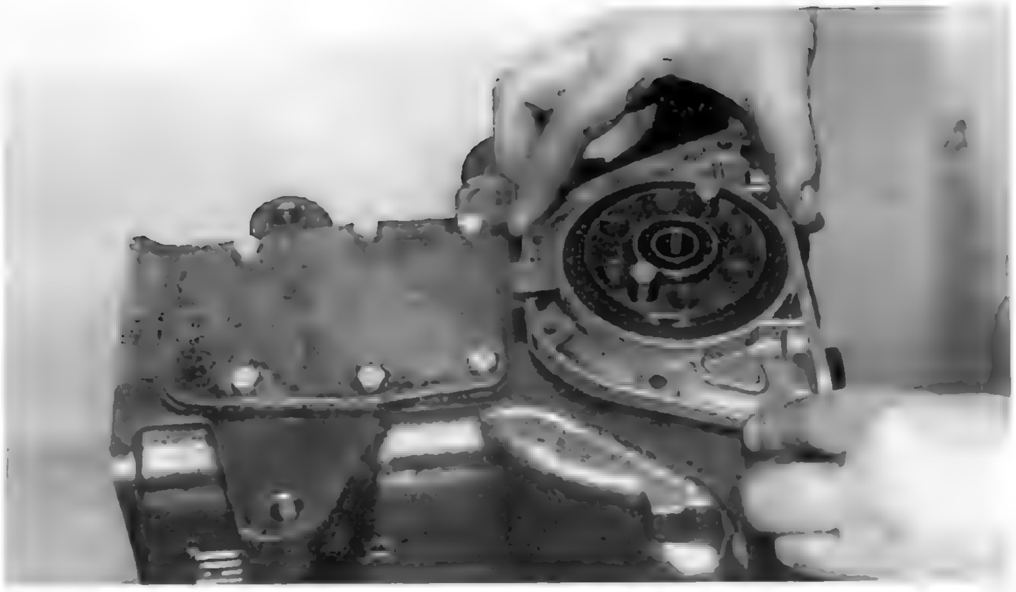
14/4: Before bolting up caps, use dial gauge with magnetic base to measure end float by levering crank to-and-fro with screwdriver. In this case float is just over 4thou". Too little and insufficient clearance may cause seizure; too much and action of clutch will cause impact damage between crank and washers. Thrust washer damage is unusual and is normally caused by neglect of oil. Thrust faces of crank can be redressed on crank grinder and oversize washers fitted. This crank is 131 2l with gearbox input shaft bearing fitted. Note use of dowels (stainless steel) for race flywheel on this Hot Rod motor.



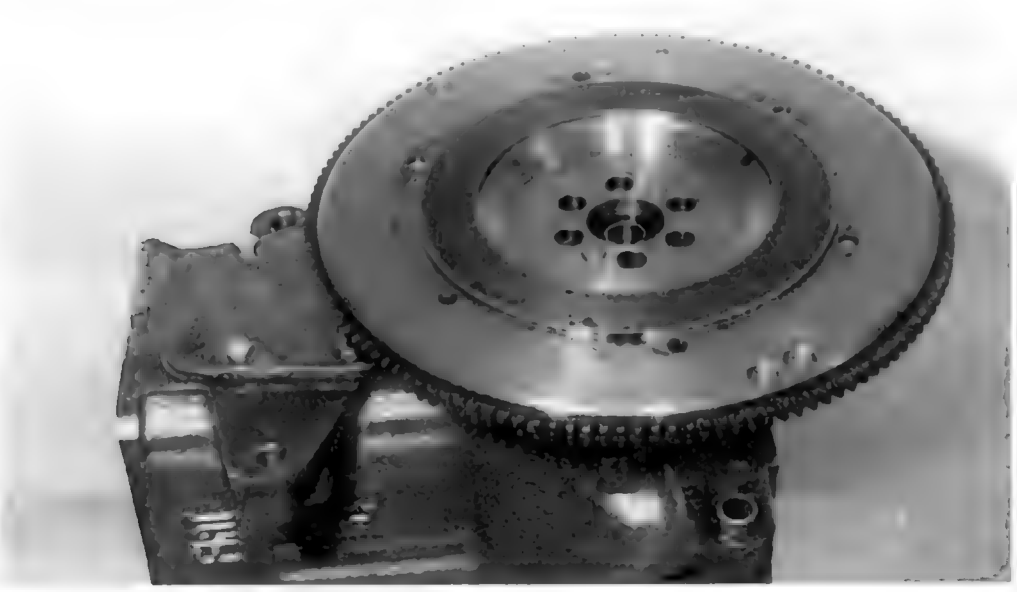
14/5: Tighten up main bearing caps in pairs with ratchet wrench and after each pair is secured check that crank rotates freely by hand (should turn easily with two fingers). Torque up only after all have been checked. Torquing up twice ensures an untroubled night's sleep! For torque accuracy, only use top-quality wrench, eg Britool, Snap-On or Facom, and check calibration each year.

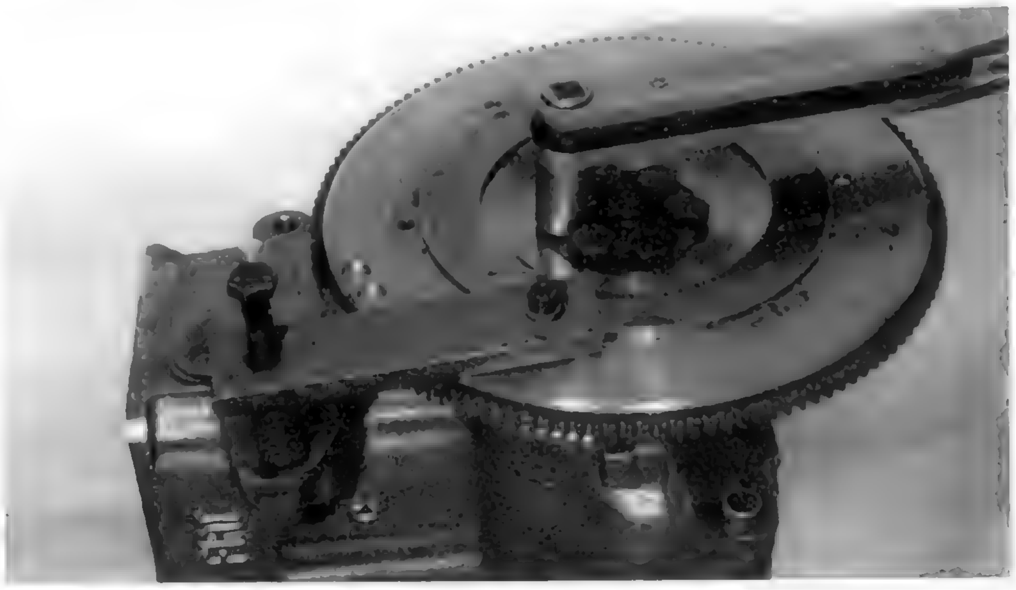


14/6: Fitting block end plate: Use of hex bolts here rather than cap heads since clearance to bellhousing can be a problem. If end plate or block is corroded, use silicon sealant with gasket. If this component leaks, whole gearbox has to come off! Note locating dowels on lower block mounting points for bellhousing. This ensures input shaft centres correctly. Commonality of mounting points is great feature of TCs.

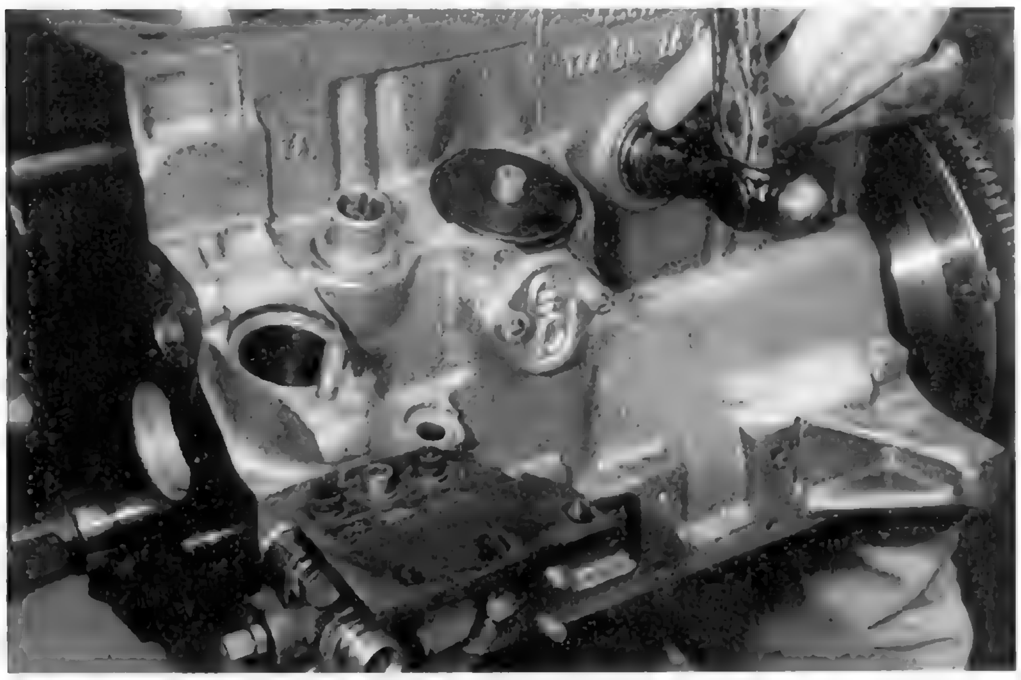


14/7: Tap new crank seal into housing with hammer, or use press if available. Lubricate with graphite grease or oil and gently ease over crank flange and bolt up with new gasket; 20mm cap heads (preferably stainless) improve appearance. On competition engines use of Loctite on all seal housings is strongly recommended, even with spring washers fitted, as they can vibrate loose.

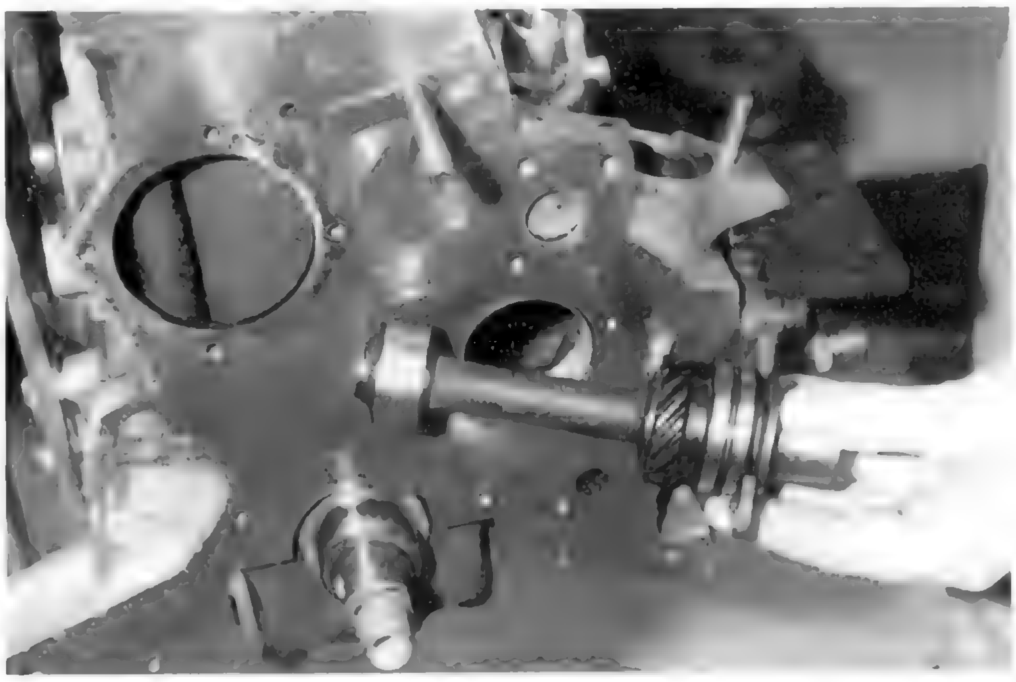




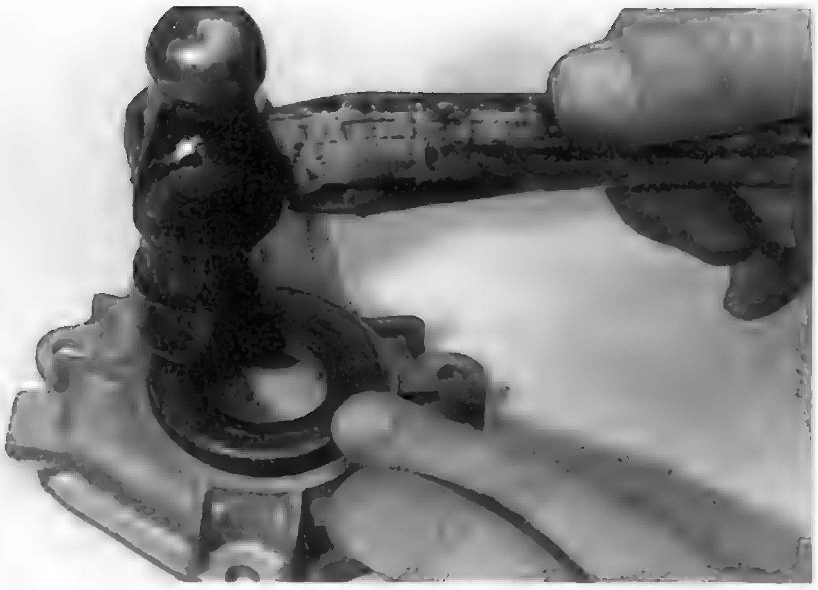
14/8, 14/9: Lock up flywheel and torque up, double-check all bolts are tight; don't forget spring steel washer. Model shown has early GC steel flywheel and double dowels. (All TC cranks are pre-drilled for dowels 10mm ⌀). Note special machining for 7¼" race clutch. In this case clutch bolts require nyloc nuts as flywheel is 'through' drilled. Balance correction holes visible. This flywheel was raced by customer for a whole season without balancing (luckily nothing broke!) but balance holes show it was some way out! Always ensure crank/flywheel are marked to show relative position after balancing. Although outer edge of flywheel in this case is only 4mm thick, centre and friction face are 8–10mm.



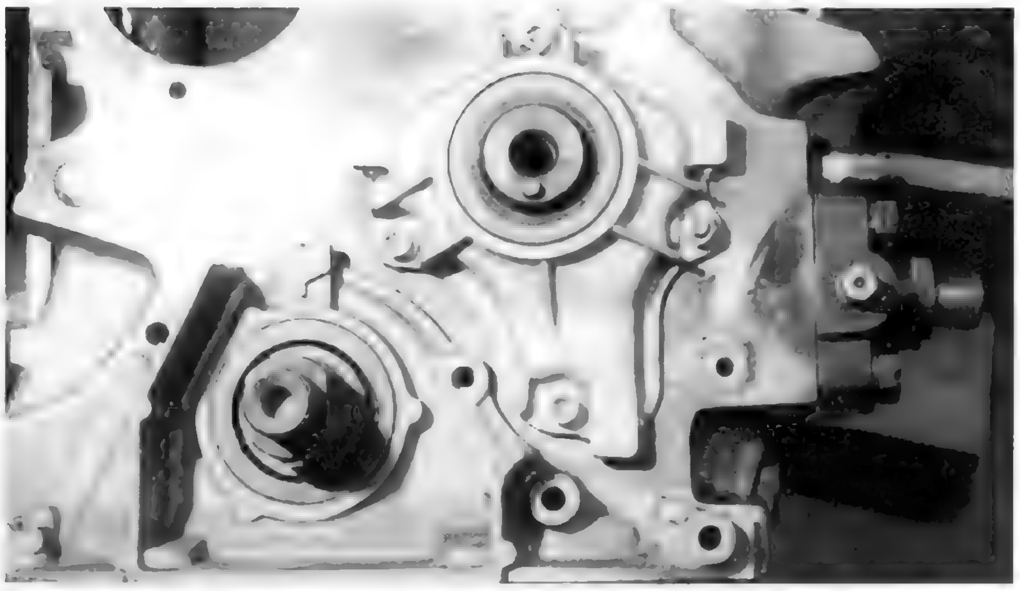
14/10: At this stage, a few ancillary items can be fitted – core plugs shown here: seal block orifice with silicon gasket and tap into place using suitable size drift or socket. Outer edge of plug should align with chamfer in bore. Forget bolting them in – a total waste of time because no way is 15lbf/in² cooling system pressure going to blow out the core plugs. Breather on this dry-sump engine runs direct to oil reservoir, co-located with engine so scavenge unit is blanked off. If tank is some distance away, standard breather system may need to be retained. Bracket on oil take-off plate is for dry-sump pump.



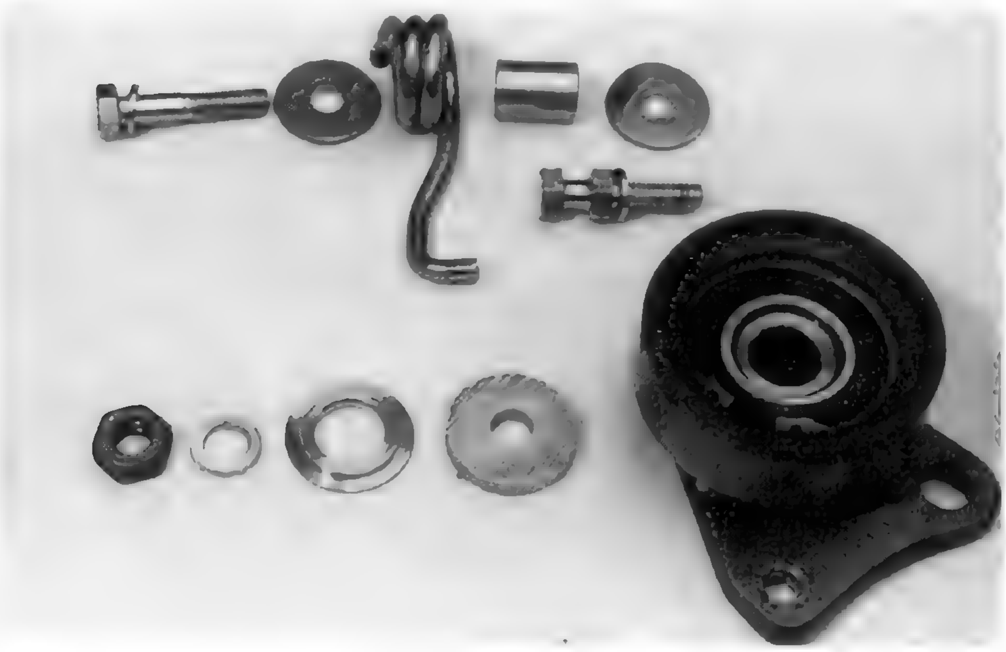
14/11: Modified auxiliary driveshaft with retaining collar. Lubricate journals with graphite grease, instal and bolt up (15mm long bolt). Core plug above driveshaft never needs to be removed. Similarly avoiding disturbing oilway gallery plugs – which should be left alone – clean galleries with high-pressure washer, airline and aerosol cleaner. If you try to replace them, they may leak. (Threaded plugs are best.)



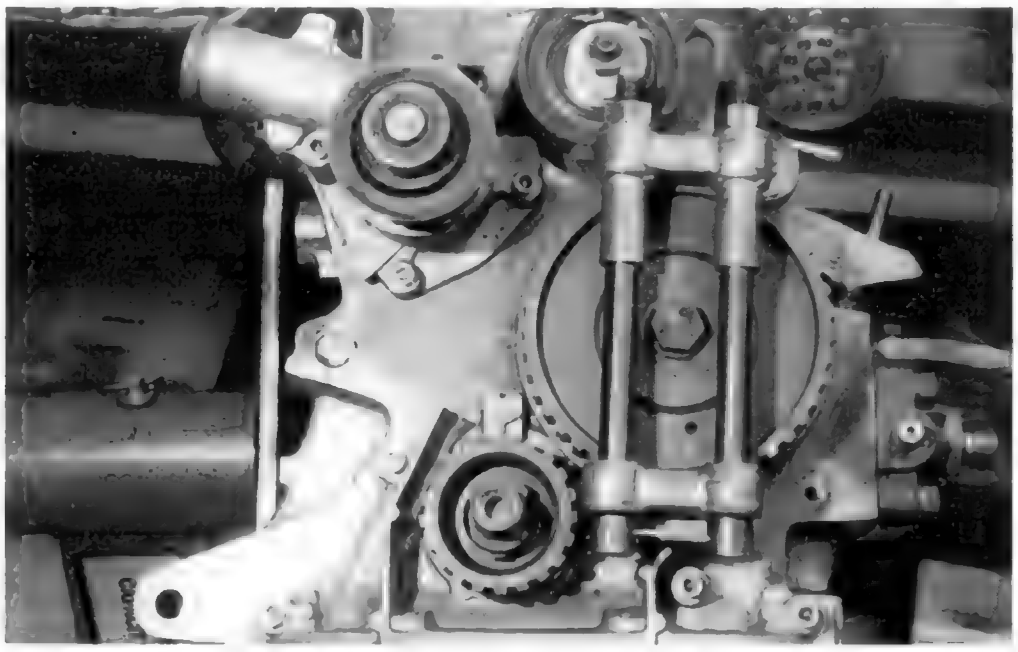
14/12: Tap auxiliary driveshaft seal into place. Late-pattern seal (eg Tipo type) has rubber seal on outer casing. If using early seal, silicon gasket in housing may prevent oil leaks. Auxiliary driveshaft seal is same on all 8v models. To avoid damage to seal casing, a steel plate can be positioned over the seal to drive it home.



14/13: Fit auxiliary driveshaft seal housing and crank front seal housing with new gaskets. Front seal housing is late type designed to shield belt from stones. On early models, eg 124 1800, front seal acts on alternator drive pulley, not crank, and a sealing O-ring must also be fitted to rear of pulley. Late models have oil pump incorporated into housing, driven off flats on crank nose.



14/14: Early tensioner components. Type of backplate and pulley varies – if swapping between models, ensure pulley/belt alignment is correct; this is a Beta 2l pulley. Note special fine thread on spring retaining bolt (top left). Stepped bolt (top right) can be replaced with standard 8.8-grade (or better) with nut attached – standard item is prone to shear. Interestingly, bright zinc-plating raises strength of fasteners by about 20%.



14/15: Do not forget new copper sealing washers on water pump – holes in block are drilled right through into coolant gallery. Some models, eg 8v Integrale, have fine-thread 8mm bolts on water pump – don't lose them! Special tool used to secure auxiliary driveshaft pulley; in this case, rear flange type for 1" belt. A large strap wrench can also be used.

DRY-BUILDING

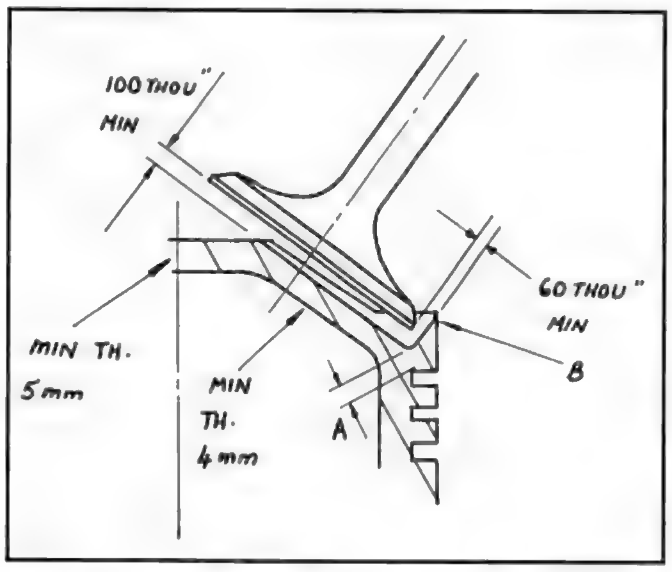
The purpose of dry-building is to ensure that the necessary vertical and radial clearances exist between the valves and pistons. Incorrect (too tight) clearances may lead to a dropped valve, where it is hit by the piston, breaking or at least bending it. The clearances derived by GCT are based on experience – deviate from them at your own risk! With 44/38 valves, the Guy Croft Formula 2000 hydroplane (1800) was run at an incredible 10,500rpm at Lake Windermere in 1989 using triple springs. [*Author's note:* The boat had a 'telltale' tachometer – you couldn't take your eyes off the water to read instruments at 98mph!] Spring control: running clearances between valve, guide and piston (particularly if worn, at the end of a season) must all be allowed for.

Vertical clearance must be allowed to permit expansion of the piston, crank flex, rod stretch and taking up bearing clearance and possible valve float (with triple springs this is negligible, unless the unit is chronically over-revved). Radial clearance must allow for piston 'slop' and guide wear.

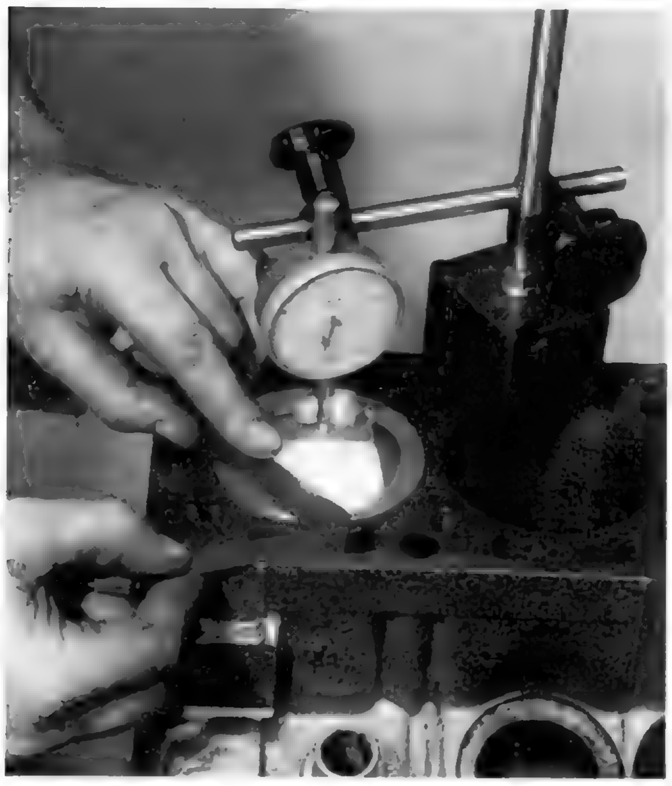
Between TDC and 20deg ATDC inlet cam profiles lift the valve faster than the piston moves away. With the exhaust the closest point occurs between 20deg BTDC and TDC. The exact position of closest proximity can be determined by inserting a feeler gauge between the valve and piston through the opposite port, but it is not really necessary to establish this unless the cutout depth is absolutely crucial to achieving a high CR (this technique was used on the Volcanic Pistons – *see Case History*).

On all the TCs, provided that only 10thou" or less is machined off the head and block and cams of 10.4mm (true) lift are used, it is not generally necessary to dry-build when using cast pistons of the various types (132 1800, 131 1600, Beta 1600) available as their cutouts are deep enough to accept the valve lift around TDC. (This is based on the piston crown being flush with the top of the bore and valve sizes up to 43½mm (inlet) and 36mm (exhaust).

One important feature of the TCs (84mm bore versions and 1,592cc only, excluding 16v) is that Nos 1 and 4 combustion chambers are offset towards the crank centre journal by 3.5mm. This resulted from the use by Fiat of the same patterns for the 1608 head castings being adapted to suit the longer 84mm block. This disparity makes accurate dry-building valid to avoid cutouts being any larger than necessary (or too small!).

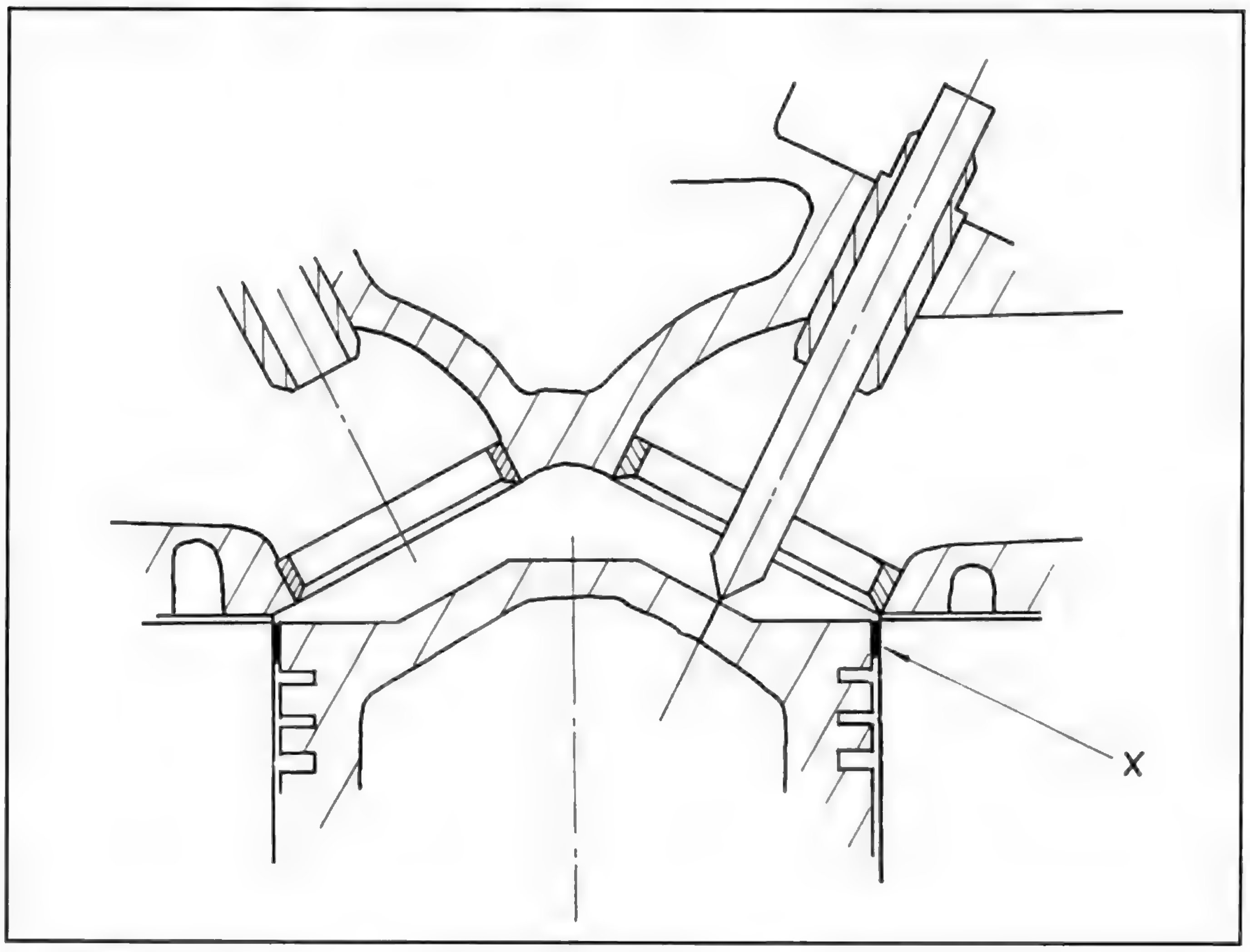


14/16: DRY-BUILDING
Vertical and radial piston-valve clearance must meet dimensions shown. Minimum crown thicknesses given are for forged pistons. Cast pistons for normally aspirated use need to be thicker; for turbo minimum of 7mm on crown depending on exact design. Do not allow cutout too close to top ring groove A – 2–3mm is about minimum for adequate strength. Cutout depth is mainly governed by valve lift at TDC. Blend cutout at B if a sharp edge results. Ensure a radius is allowed at base of cutout as shown.

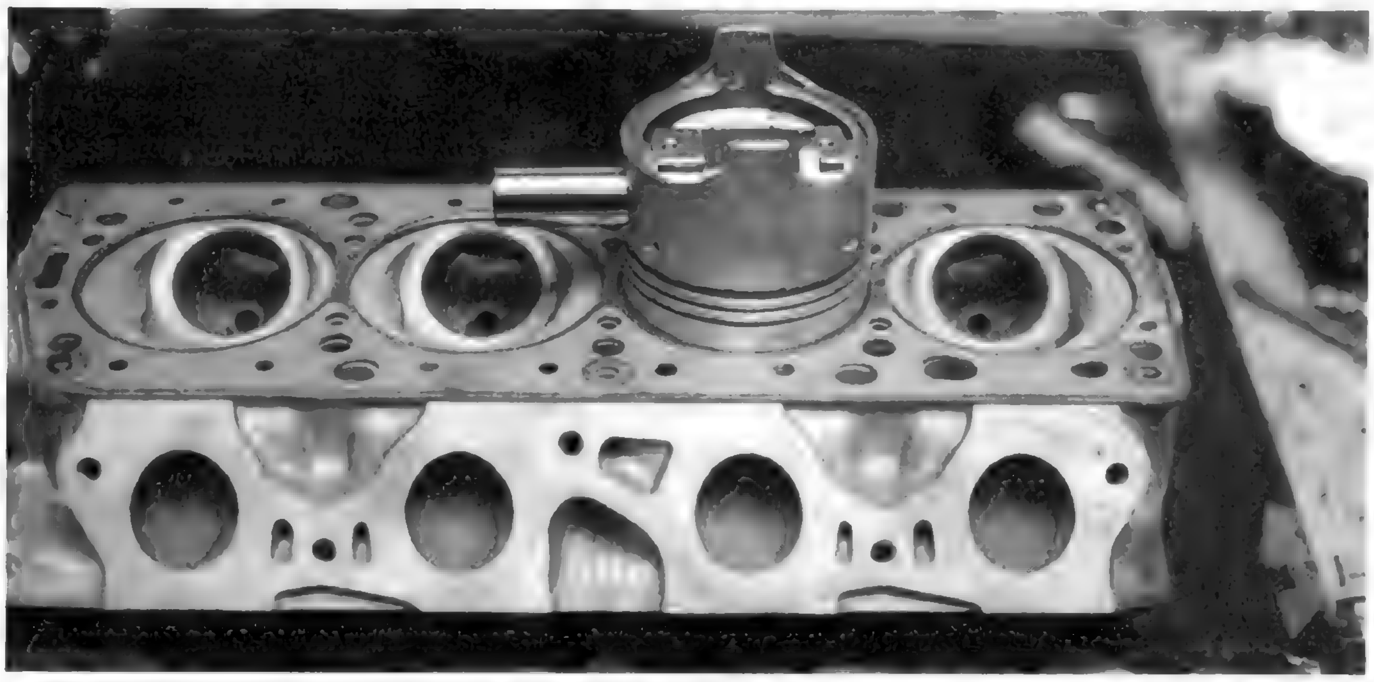


14/17: At this stage, and before con-rod caps are fitted, measure the set-up height of piston crowns above block face at TDC. Pistons can be safely run up through gasket at least 20thou", but always ensure piston-head clearance is adequate. Bore of any production head gasket will only allow piston of up to 85mm (late engines) to protrude above block face. If pistons any larger, eg 86mm, are used, chamfer edges of piston crown so as not to clip fire ring of gasket. When measuring protrusion of pistons, tilt piston and assess highest point with straight edge and feeler gauge as shown. This ensures adequate safety factor.

Having established that a dry-build is needed, the first job is to mark the valve centres on the pistons as shown. (Do not build up the head at this stage.) This technique allows the milling machine to accurately position on the piston in the exact valve centre for flycutting the valve recess (cutout).(14/18, 14/19)



14/18: Using old valve stem (8mm) with sharp point to pin-punch valve centres on pistons prior to machining cutouts. Head is clamped in place with two head bolts, with gasket and locating dowels in place. Valve head angles on all TCs (early and late port layout) are inlet 31° 45', exhaust 33° 30' to horizontal. Packing at X keeps piston in place.



14/19: Quick way to mark pistons without stripping down head. Lay piston centralized in gasket bore, as shown, and with pin parallel to crank axis, hold piston firmly and, with punch in valve guide, tap it gently with hammer to mark valve centre. Remember to get cutouts right way round and number pistons! As this method is not as accurate as doing it with pistons in their bores, allow an extra amount of radial/vertical clearance (eg ½mm).

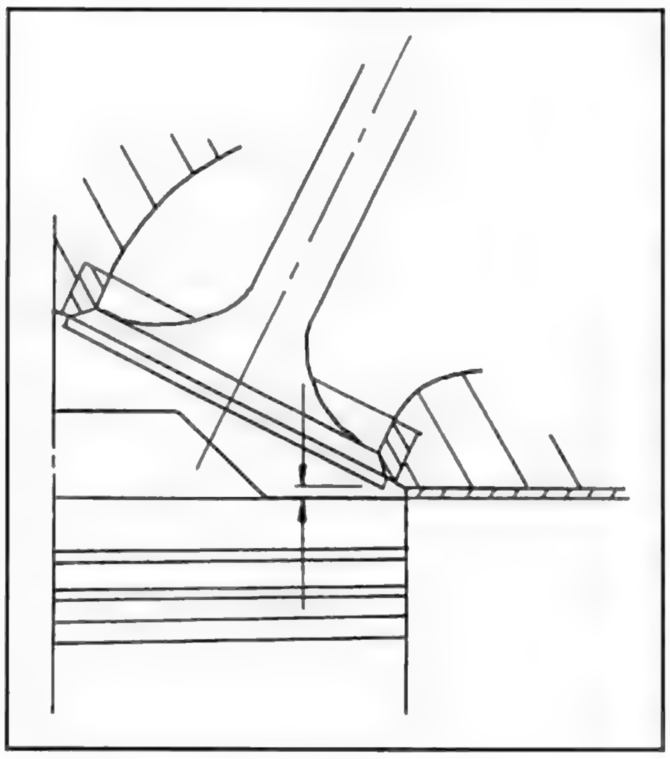
Next it is necessary to estimate the vertical clearance required. This will be determined by:

1 Crown/dome height of piston at TDC.
2 Valve lift at TDC and accelerative profile of cam.
3 Gasket thickness.
4 Available metal in piston dome/crown for removal.
5 Position of valve relative to head face when closed.

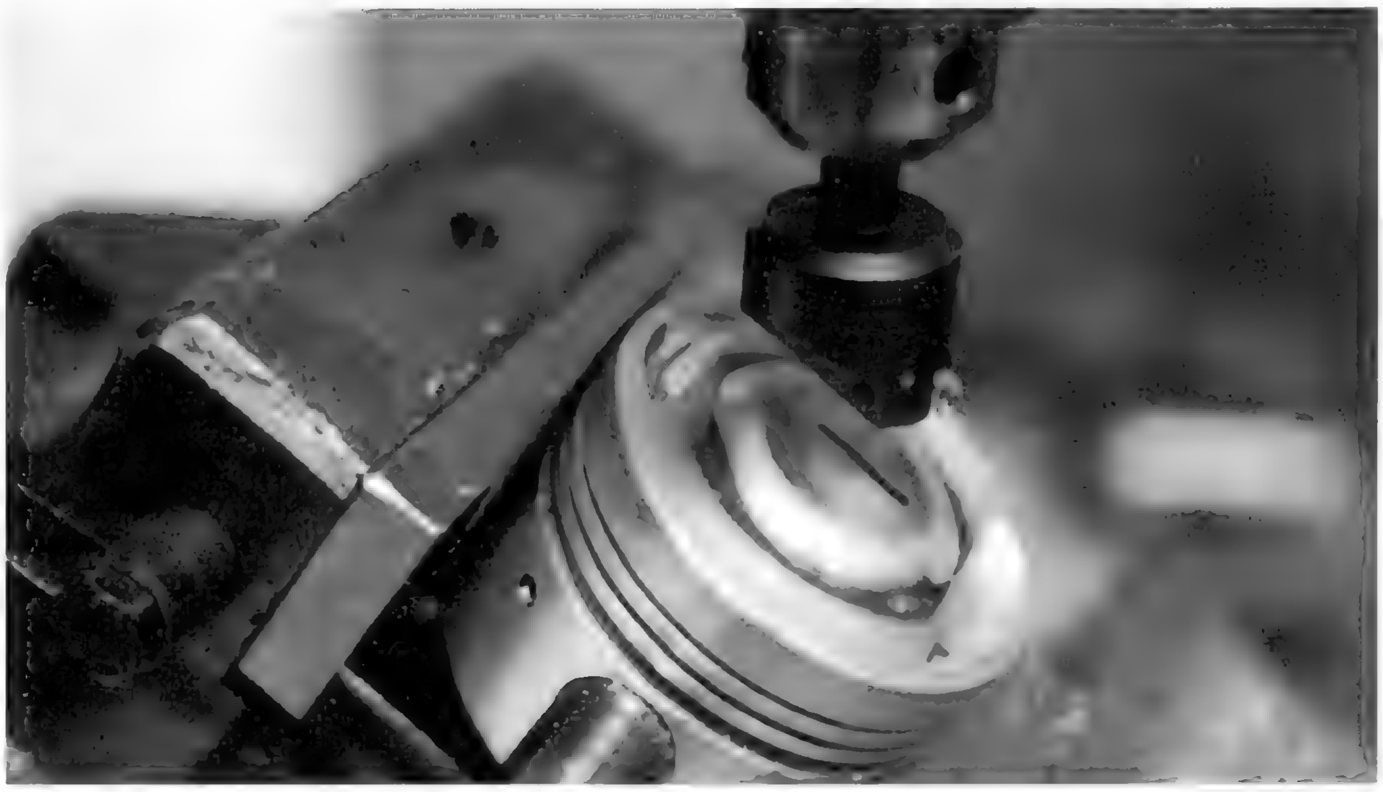
(See diagram 14/20.)

If no preliminary clearance exists (because the crown or dome interferes with the valve) some initial matching must be carried out. This is easy enough: with the pistons inclined on the mill bed and the flycutter centred correctly over the valve stem centre (pin-punched earlier), the cutter ⌀ can be set to valve dia +120thou" (2 × radial clearance of 60thou") and a light preliminary cut made depending on the degree of interference.

In the diagram shown, the inlet valve edge is already 1mm away from the



14/20: Assessing valve cutout vertical depth required. Dimension arrowed must be established – clearance between valve (inlet or exhaust) – closed – and piston at TDC.



14/21: Machining valve recess with single-point cutter. Tool is centred over pin-punch mark and cutter set to valve diameter + 2 × radial clearance. Tool tip must be radiused or cutout will be weakened.

crown. If the lift at TDC is 4mm, 3mm plus clearance must be machined out of the piston. This will need to be 2½mm, plus an additional safety allowance of 1½–2mm depending on the cam type, but usually this is determined by trial and error. (Of course, the cam lift can be plotted using a dti and protractor for this calculation.) Total cutout depth from the initial closest point between valve and piston will thus need to be 7–8mm. Normally, exhaust cutouts need to be less deep than inlets because the smaller diameter seat inserts are fitted further away from the head face.

Using a single-point tool (flycutter) is the safest way to machine the cutouts (**14/21**). Ensure the mill is horizontal and cross feeds are locked so that the risk of the piston becoming displaced is minimized. Use of a large slot drill speeds the process, but the stresses involved are great and the operation must be carried out most carefully.

Radial clearance check

Place Nos 1 and 4 pistons loosely (less rings) in their bores with the crank at TDC. Place Plasticine in the cutouts. It is worth packing the piston on the side being measured to ensure that the 'worst-case' radial clearance can be assessed. Fit the locating dowels, two dummy head bolts and the gasket (preferably a used one) and with two of the valves in position (no springs fitted) and their heads greased to prevent the Plasticine sticking, lay the head carefully in place. Press the valves into the Plasticine so that when the head is removed, the impression of the valve is left in the valve recess; by slitting this with a sharp blade, the clearance between the edge of the valve and the cutout can be measured (either with a feeler gauge or by peeling it off and measuring with a calliper). Then do the same with either No 2 or No 3 piston (the cutout positions are symmetrical because the combustion chambers are not offset.

If the radial clearances (inlet and exhaust) are less than 50–60thou", they should be remachined. If they are correct, check at this time the clearance between the dome and combustion chamber, using Plasticine again; a minimum of 25thou" must be allowed. If it is too close, it may knock in service; this sounds like a damaged bearing and is most alarming! The chamber can be relieved quite safely with a die grinder. Usually a clearance problem here only occurs on the front of No 1 dome or the back of No 4.

Vertical clearance check

The head may be built up if the radial clearances are OK. Dial in the cams, shim-up the clearances and install all the pistons (with or without rings) and tighten up the con-rod bolts (it is not necessary to torque them at this stage – but don't forget to do it later!). Place Plasticine in cutouts 1, 2 and 4 (about ⅛" strip will do), grease the valve heads and bolt the head in place. If an old gasket is used, no allowance needs to be made for the compression of the gasket.

Fit the cam belt correctly and rotate the engine two complete revolutions. Remove the head and measure the depth of the compressed Plasticine by slitting it with a blade and use of a feeler gauge or calliper. If the vertical clearance is wrong, the cutouts must be remachined. To save time, a spare bare head is useful for re-marking the valve centres! Keep a record of the measurements; it is not enough to say "they are all OK", you need to know exactly what they are. This, apart from anything else, enables you to make a cam swap at a later date.

> **REDUCED CLEARANCE**
> (Owing to the inclination of the valves to the vertical, if the vertical heights of gasket, block face or head are reduced, the radial valve-piston clearance is reduced by approximately a third of this amount, *ie* if 18thou" is machined off the block, the radial clearance is reduced by about 6thou".)

Typical table of valve-piston clearances:

Cylinder		*inlet* (mm)	*exhaust* (mm)
1	vertical	2.5	3
	radial	1.5	1.5
2	vertical	2.5	3
	radial	1.5	1.5
3	vertical	2.5	3
	radial	1.5	1.5
4	vertical	2.5	3
	radial	1.5	1.5

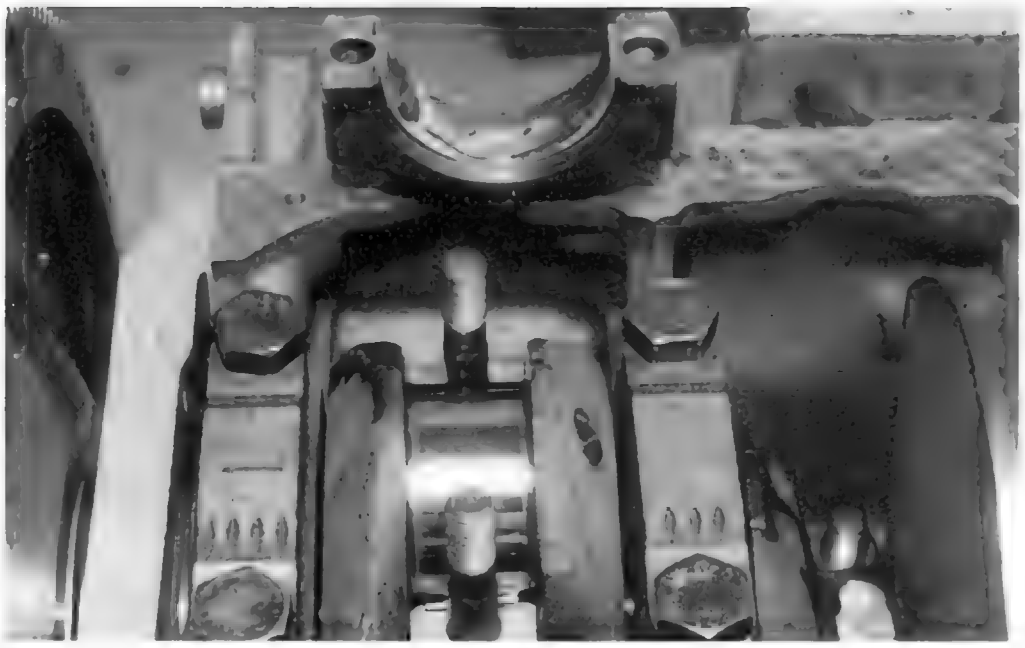
Allowance: at closest point, minimum
vertical clearance – 100thou" (2.5mm)
radial – 60thou" (1.5mm)

If the dry-build clearances are OK, the engine build-up can progress.

Note: GCT have had to dry-build some engines three times. This is most tiresome, but worth the effort at the end of the day because the engine can be run flat-out with confidence.

14/22: Lubricate bores and pistons. Next, install pistons using ring compressor. Chamfer around bores helps to ease entry. When tightening this type of tool, make sure tool does not catch on ring ends. Positioning con-rod centrally on pin and keeping piston properly aligned will prevent rod fouling crank. Tap piston firmly with hammer handle as shown to engage rings in bore, then smoothly push down until bearing engages on crank.

14/24: Oil bolt threads and tighten all nuts spanner-tight to prevent bearings being disturbed, then check crank rotation before torquing-up. Next torque-up nuts progressively and check crank rotation again. Finally, double-check for tightness and rotate engine one complete revolution to make sure it turns without binding. Tightness is caused by:
1 – bent crank (or badly ground)
2 – dirt under bearing (or wrong bearings)
3 – distorted housing



14/23: Carefully fit caps to rods. Ensure numbers of caps marry up and if using original crank and rods, re-install them same way round, ie numbers on left (viewed from crank nose); this will ensure wear patterns on crank and rods will match, but this is not crucial. Early (124 1800) rods must go in correct way round due to position of oil spray holes.

With early 2l, press-fit bolts will not fall out with engine upside-down as shown; not so with smaller models, so position engine accordingly. Similarly, 2l Beta/131 types are fairly spacious and nuts can be fitted on all rods whether pistons are at TDC or BDC as shown. On non-press fit models, fit nuts to Nos 1 and 4, then rotate crank to gain access to Nos 2 and 3. Make sure, on models without press-fit rod bolts, that bolt heads do not become unseated during tightening. Block numbers on main bearing caps clearly visible in this shot. Late models have bolt-only fixing on rods.

TURNING TORQUE

The turning torque of an engine is essentially the torque required to turn it. This can be measured simply with a torque wrench. On new GCT engines (cold, with only preliminary lubrication) the following may be a useful guide:

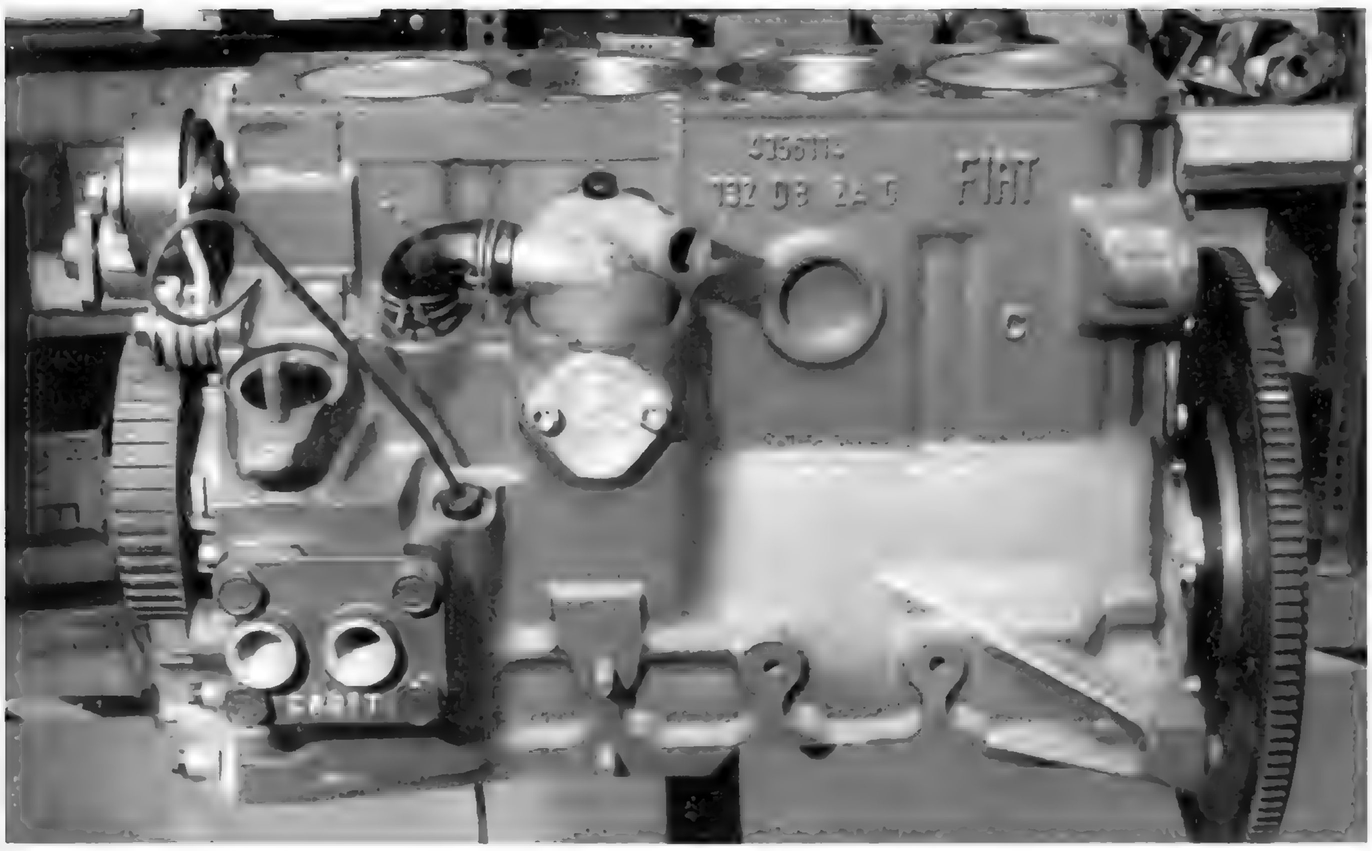
2*l* block assembly – 12lbf ft
2*l* build engine – 24lbf ft (St III cams, plugs removed, triple springs)

The turning torque of a run-in engine (not firing) is less, of course, but when the engine is firing, under load, additional torque is required to turn the oil pump and water pump, and the ring friction is much greater due to the firing pressure. Added to this there is the 'pumping loss' on intake and exhaust to take into account. Neglecting these effects, however, and applying the equation:

$$\text{Power (bhp)} = \frac{\text{Torque (lbf ft)} \times \text{Speed (rpm)}}{5250}$$

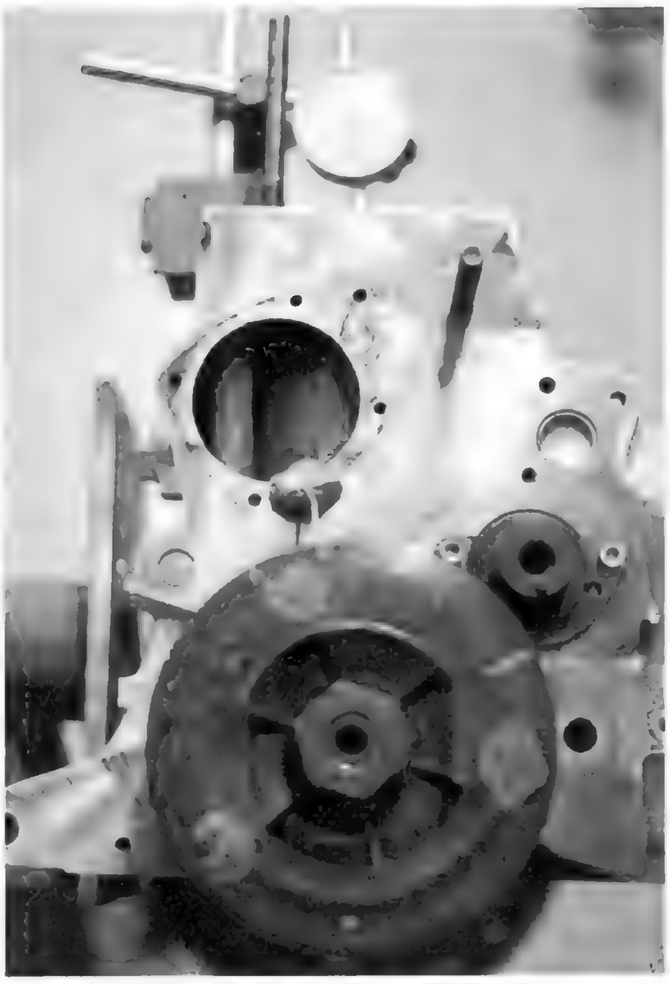
gives a power requirement of 25bhp just to turn the engine at 7000rpm. In real terms, the St III engine (with 70% ME at 7000rpm) loses 52.5bhp.

[*Author's note:* Some people refer to a 'loose', older engine as giving more power. GCT dyno tests disprove this: the ME of a TC 'run-in' after 2½ hours on the dyno gives almost exactly the same ME tested 'as-is' (*ie*, taken out of the car and put straight on the dyno) at the end of a hard season's racing. If you want minimum power-loss, build it right first time.]

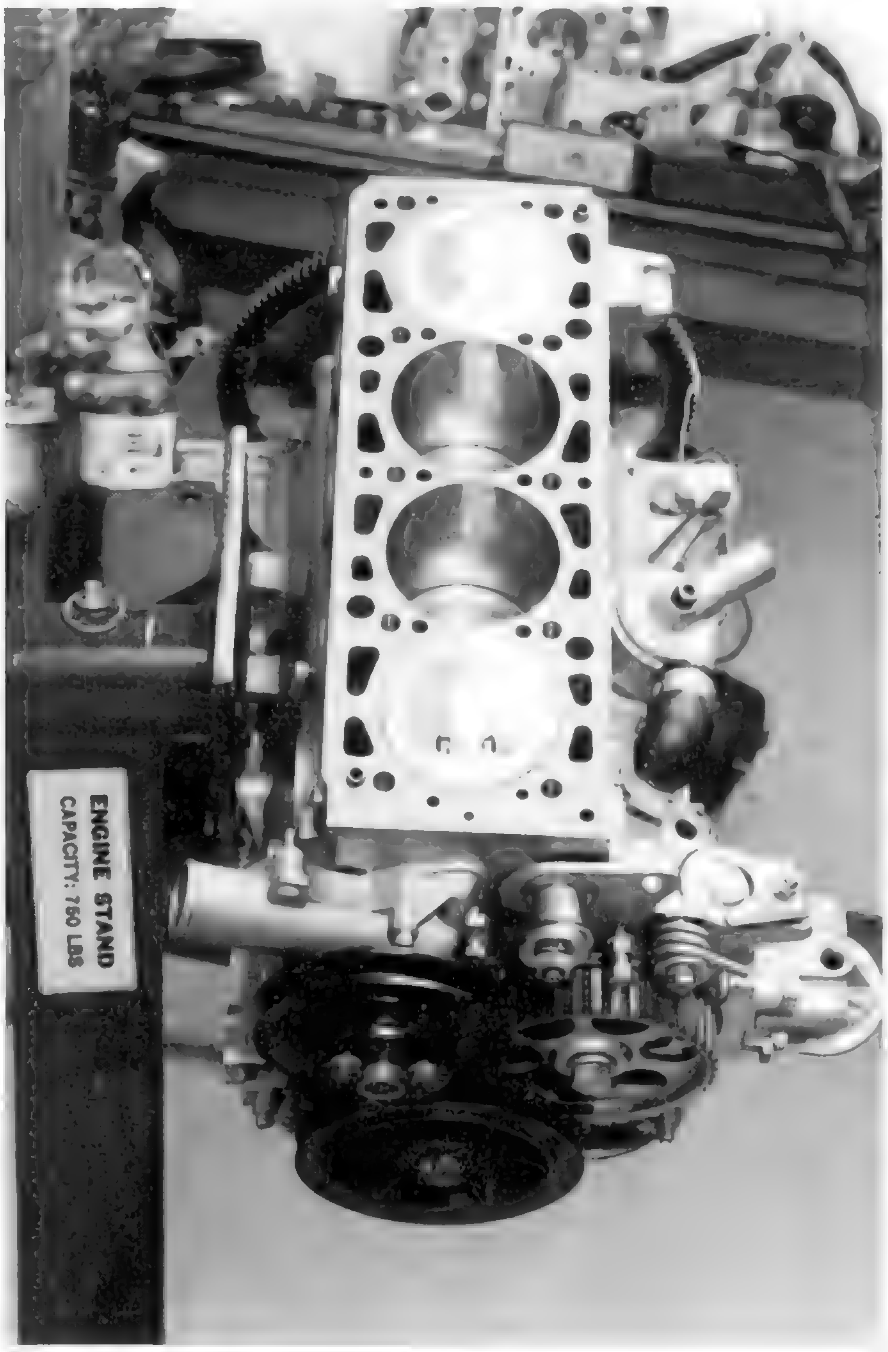


14/25: Built-up 131 2l block with scavenge breather unit fitted. Unit returns oil to sump via tube and breathes gas to air cleaner on standard engine; 130 TC bleeds gas to special tapping on carbs. Most race engines run breather into catchtank. Standard breather quite satisfactory up to St III, but for full-race applications function is better performed by scavenge rotors on dry-sump pump, due to extra blow-by past pistons with high compression.

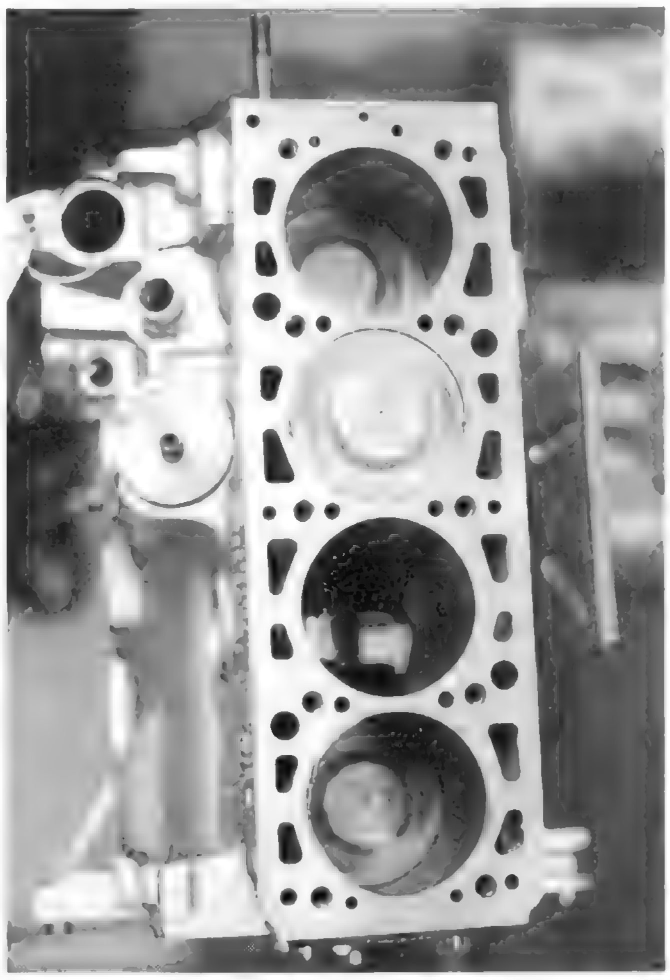
Breathers have become increasingly sophisticated (eg Integrale), but basic function remains same – to allow crankcase gases to escape. Over-pressurization of crankcase leads to oil leaks on crank, cam seals and sump gasket, not to mention high power loss. On that subject, it is important that hose to catchtank is at least same bore as breather port. Note that this engine has standard compression ratio flat-top pistons.



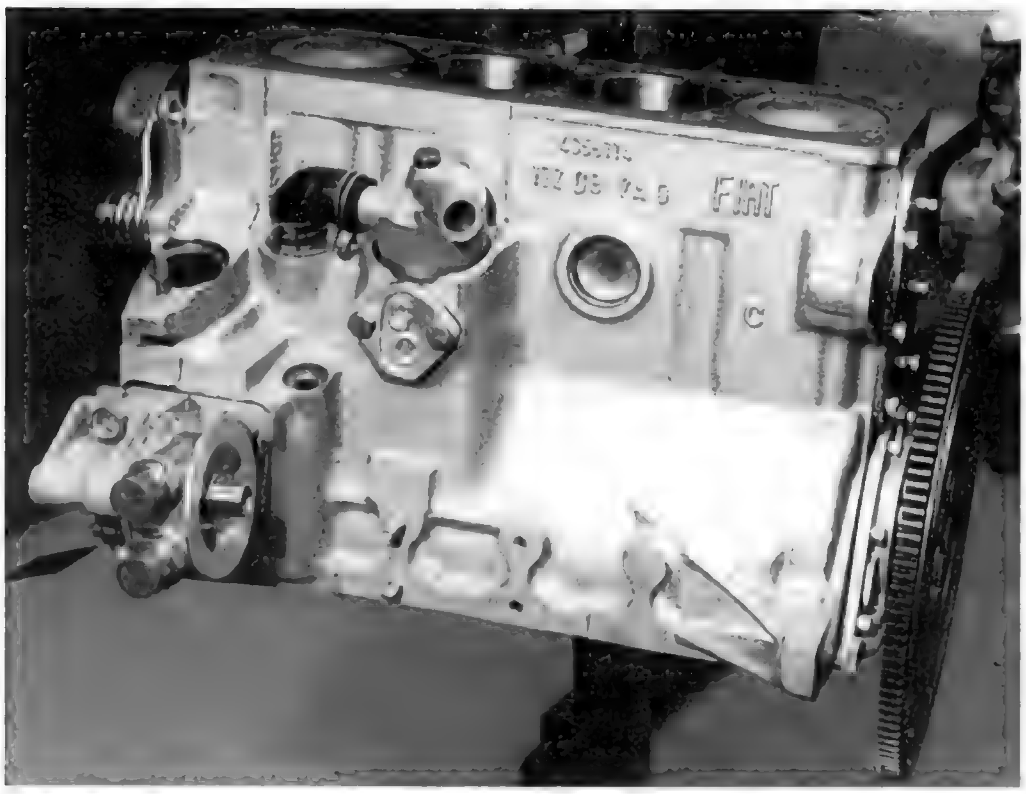
14/26: ESTABLISHING TRUE TDC.
Find approximate TDC using dti on No 1 piston and fit protractor and pointer to crank as shown (or to flywheel). Set dti to 0. Lower piston approximately 0.3mm and set pointer to read 0°, then turn crank other way until dti reads 0.3 again and note degree reading. True TDC is halfway between 0° and final degree reading; eg, if final reading is 22°, true TDC is at 11°. This is done to establish centre of dwell period around TDC where crank rotates but piston is stationary. This is particularly important if competition cams are being used. If true TDC is not established, cam timing will be miles out.



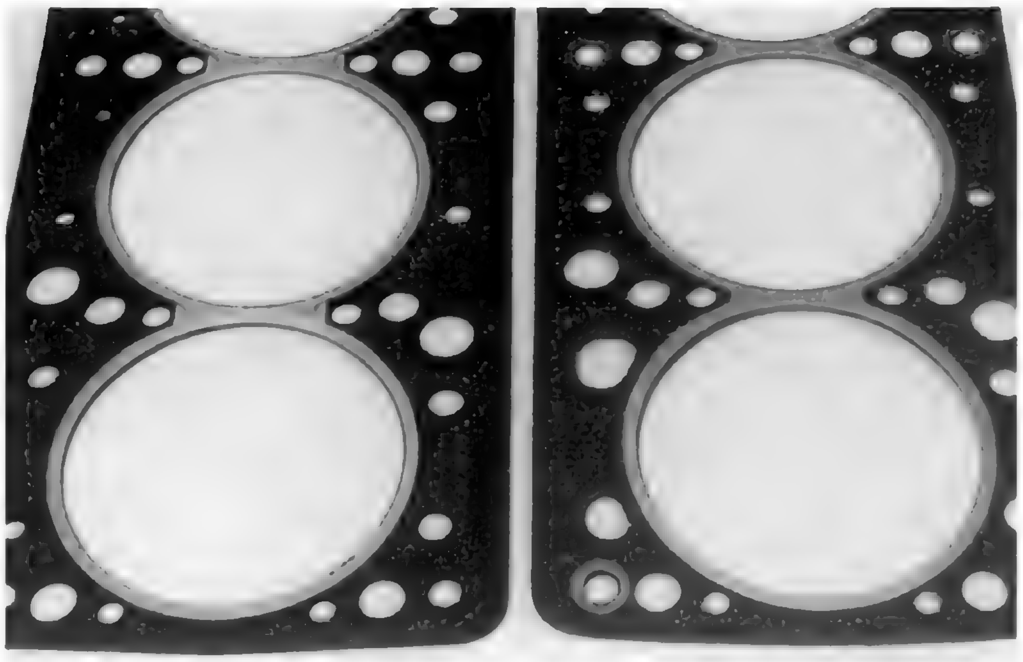
14/27: Former 130 TC unit converted to rear-wheel drive for race Morgan. Note 130 TC lightweight (sintered steel) auxiliary driveshaft pulley and early water pump and pulley. Engine has forged high-compression pistons giving approx 11:1 ratio. Flywheel pointer is affixed to block (right rear) with Plasticine – OK if it doesn't get dislodged, but bolting pointer to bellhousing mounting point is better. Engine was due to be fitted with distributorless ignition (fully mapped) hence blanked-off mounting point on block.



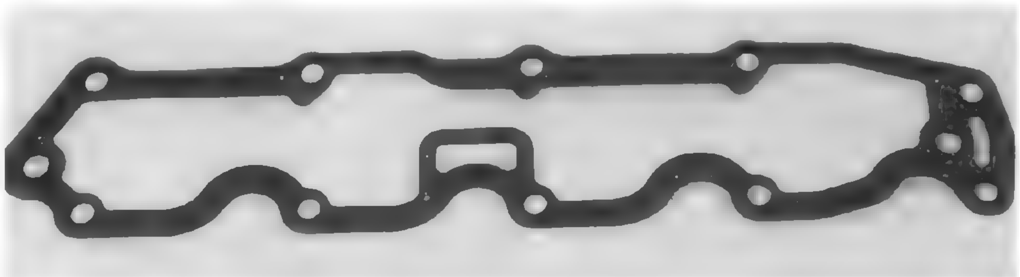
14/28: 131 2l block fitted with 131 1600 pistons to give approx 10:1 CR; breather blank-plate identifies this as a dry-sump motor where tank is co-located with the engine. Gas and oil vapour will separate in the dry-sump tank.



14/29: 131 2l block fitted with ex-Lancia Beta 1600 pistons to give around 10.5:1 CR. Oil filter housing is ex-130 TC. Standard breather set-up shown must be retained in systems where dry-sump tank is mounted remote from engine.



14/30: A tale of two gaskets: right, old-pattern Fiat/Lancia 84mm-bore gasket as made by Tako, Payen and others. Needs retorque; left, latest-pattern Fiat 130 TC-type heavily reinforced, introduced about 1992 as replacement for Goetze Astadur and comparable in every way. Fits all 84mm-bore early TCs including Volumex, Delta turbo, although later reversed-port design, eg Delta Turbo 1600 ie, Croma Turbo ie 8v or 16v, have different coolant hole layout. This type of gasket contains a polymer which hardens when heated – gasket does not settle like conventional types and no retorque is required. Fiat 'torque-angle' bolts or GC race bolts should be used with this gasket. If early bolts (hexagonal-head straight torque sequence type) are used, they must be retorqued with engine cold after 300–500 miles and if dyno-testing, before power runs. Fiat torque-angle and GC 12.9-grade race bolts can be used four times – beware of bolt-stretch on old ones, they don't last forever.



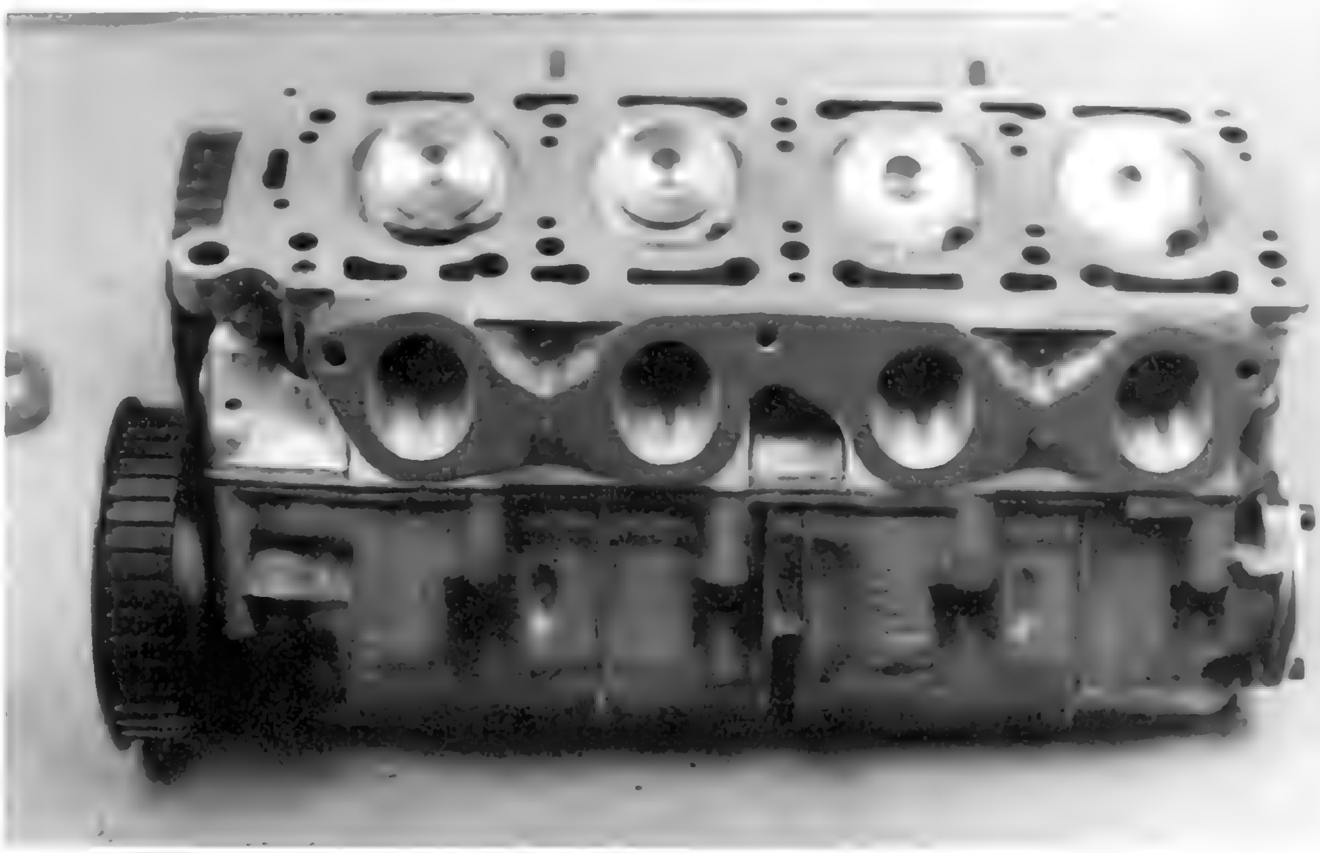
14/31: Latest-pattern cam box gasket from AE was developed after reports from GCT of serious oil leaks on non-asbestos types adopted around 1994. As AE make these gaskets for Fiat, this was a serious problem. Analysis showed tiny fissures in non-asbestos material and delamination, ie paper breaking away from stainless steel core. To their great credit, AE quickly redesigned, using non-porous paper with rubber sealing ring around outer periphery of gasket; cured problem immediately. For engines with valve clearance problems, gaskets can be stacked up to three high, but allow them to settle overnight and retorque before shimming; single gaskets bed down about 2thou". If the Fiat OE gasket set you buy contains these early gaskets, replace with new AE ones. (Or check with GCT. The sets for the reversed-port heads have the latest composition. The problem is mainly with the early engines.)

Oil pump

At this point it is a good idea to fit the oil pump (late types are fitted with the crank front seal housing) because, with the belt off, the auxiliary driveshaft pulley can be turned to check that the pump turns freely. There is an alignment tool available from Fiat to ensure the pump shaft is correctly centred in its bush, but this is not at all vital. Prime the pump gears with graphite grease and oil, and offer up to the block. Beta types use an O-ring to seal the oil gallery between pump and block; all others use a gasket.

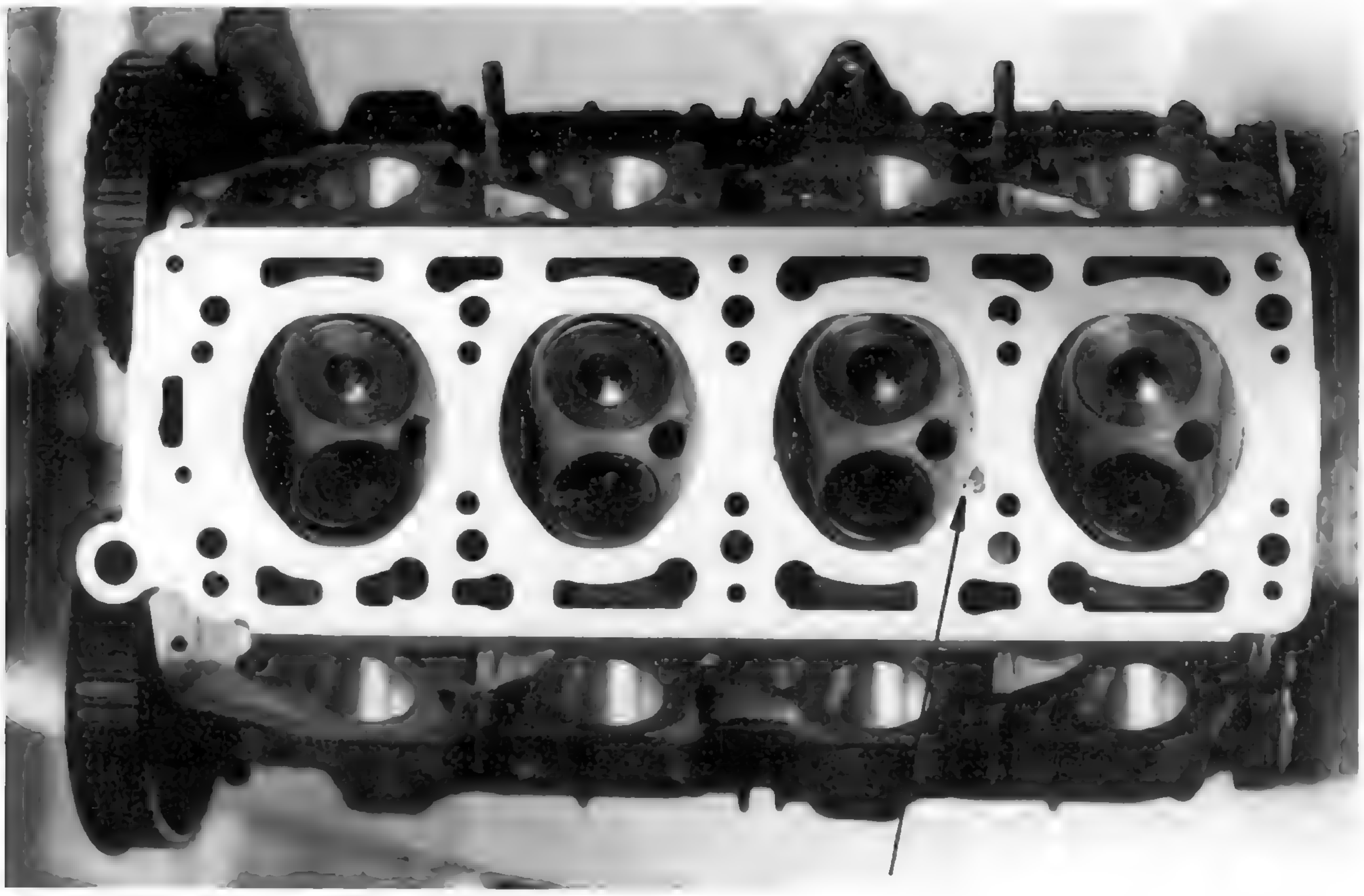
Make sure the oil pump casing bolts are secure and that the oil pump 8mm bolts have sufficient penetration into the block threads (with spring washers fitted) and bolt-up with the driven gear (a skew gear driven off auxiliary driveshaft) *in place*. Turn the auxiliary driveshaft to ensure the pump turns freely. Some non-OE pumps can be tight because they have insufficient end float, so it may be necessary to dress the gears with 220-grade emery paper to relieve them. Fit the oil pump pick-up on crank nose-driven types and the scavenge return pipe. (If a race sump is used, it may be necessary to shorten this – allow about ¾" clearance from the sump baffle.)

After fitting the skew gear, fit the retaining plug (130 TC, Volumex etc, and other models with early pump/end-drive or top-mounted distributor) and clamp, or fit the distributor to prevent the gear falling out later when the engine is inverted to fit the sump.

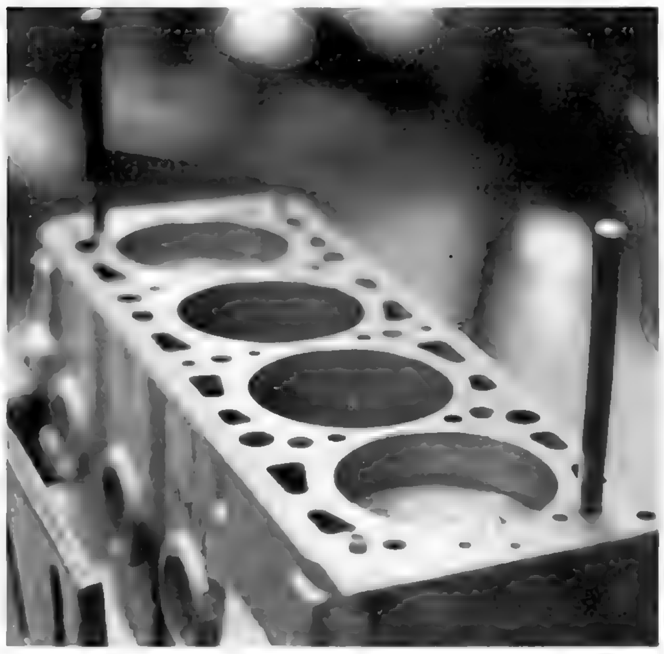


14/32: Fully ported/blueprinted Lancia 2l head ready for fitting. Note ¾" pat camwheels with flanges.

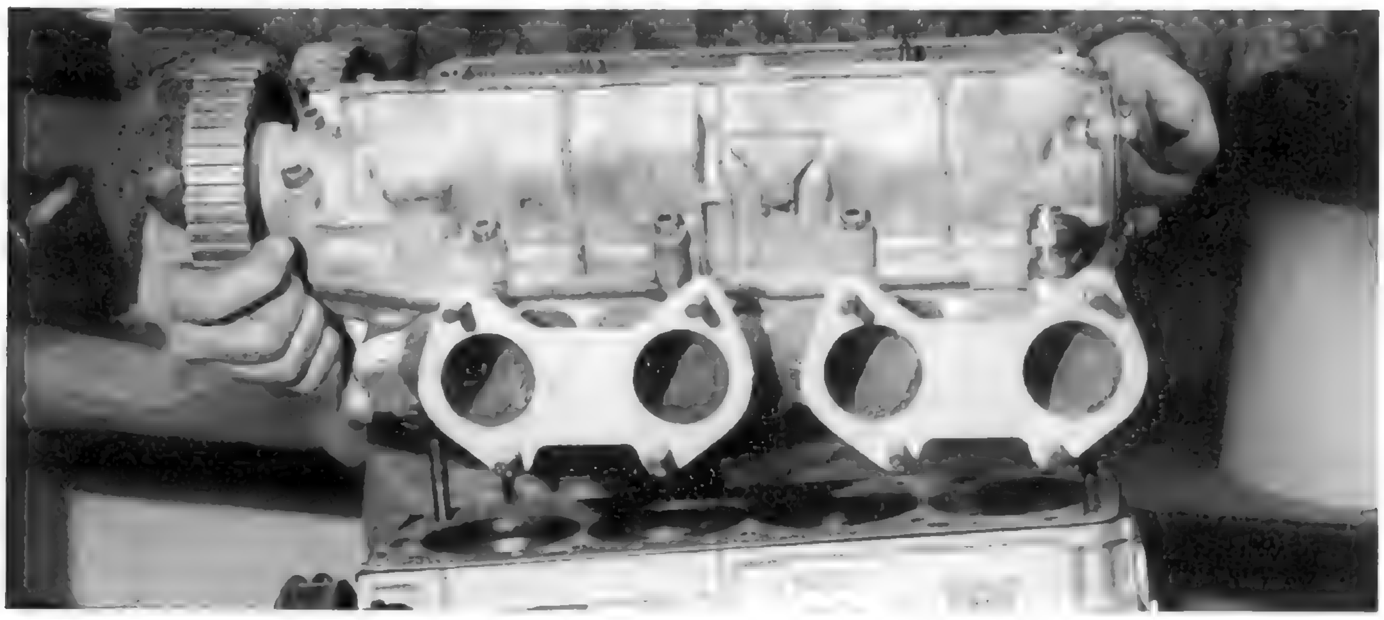
Putting on the head



14/33: Damage to squish bank (arrowed) is not significant as it is inside fire ring.

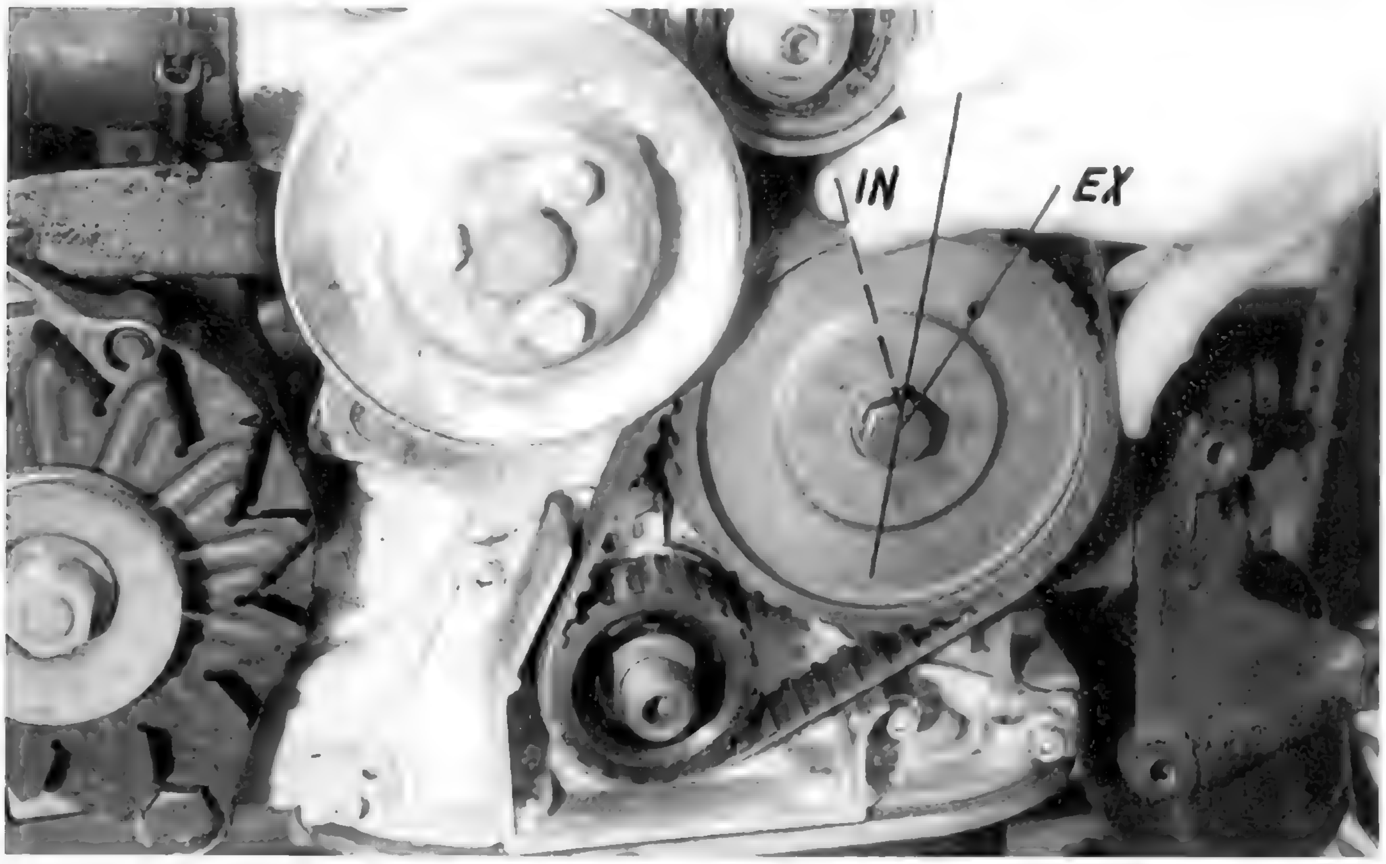


14/34: Dummy head bolts in position and gasket locating dowels ready for gasket and head to go on. Measure dowels before fitting head to ensure they locate properly in head without bottoming out; if dowels are not used, gasket will blow. Dowels for 130 TC (with late gasket – 11thou" thicker) are longer than early 124/131 types. Leave pistons ½" low at this stage, but make sure you have a TDC pointer to refer to later when belt goes on.

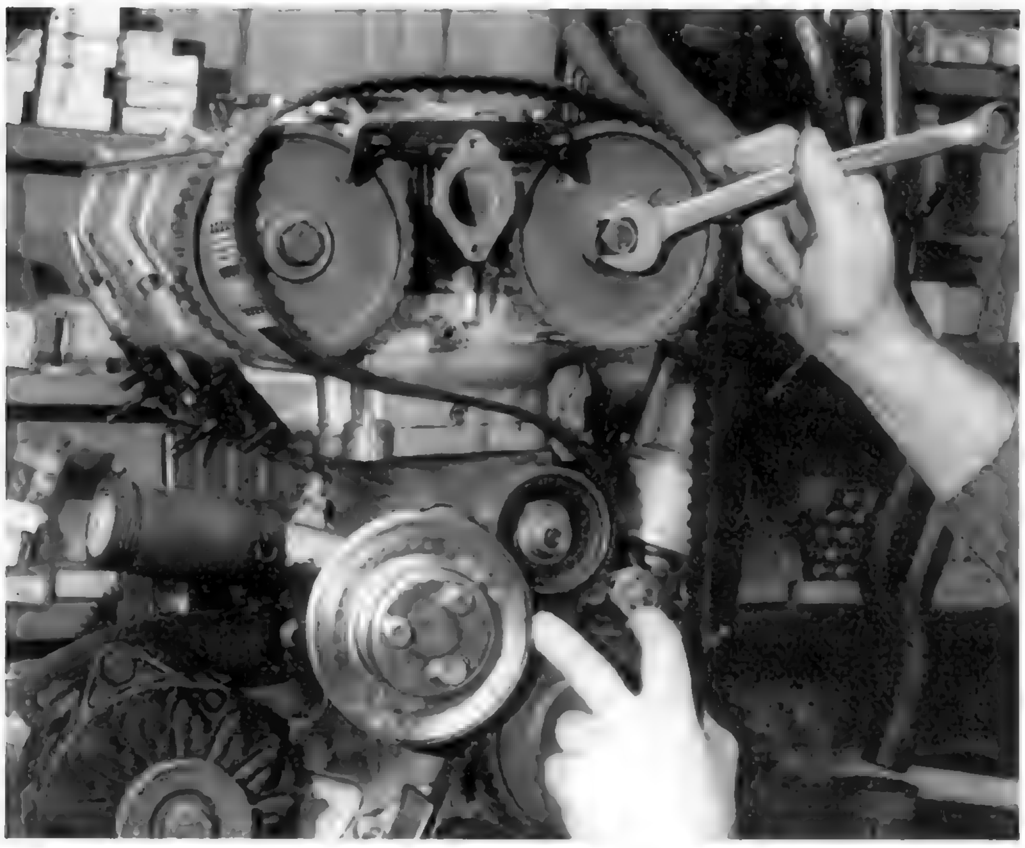


14/35: Lower head carefully into position. If it doesn't fit over dowels first time, gently reposition it – being careful not to score gasket or head face. Cams are in their TDC position, ie inlet opening and exhaust closing. Torque-up head bolts.

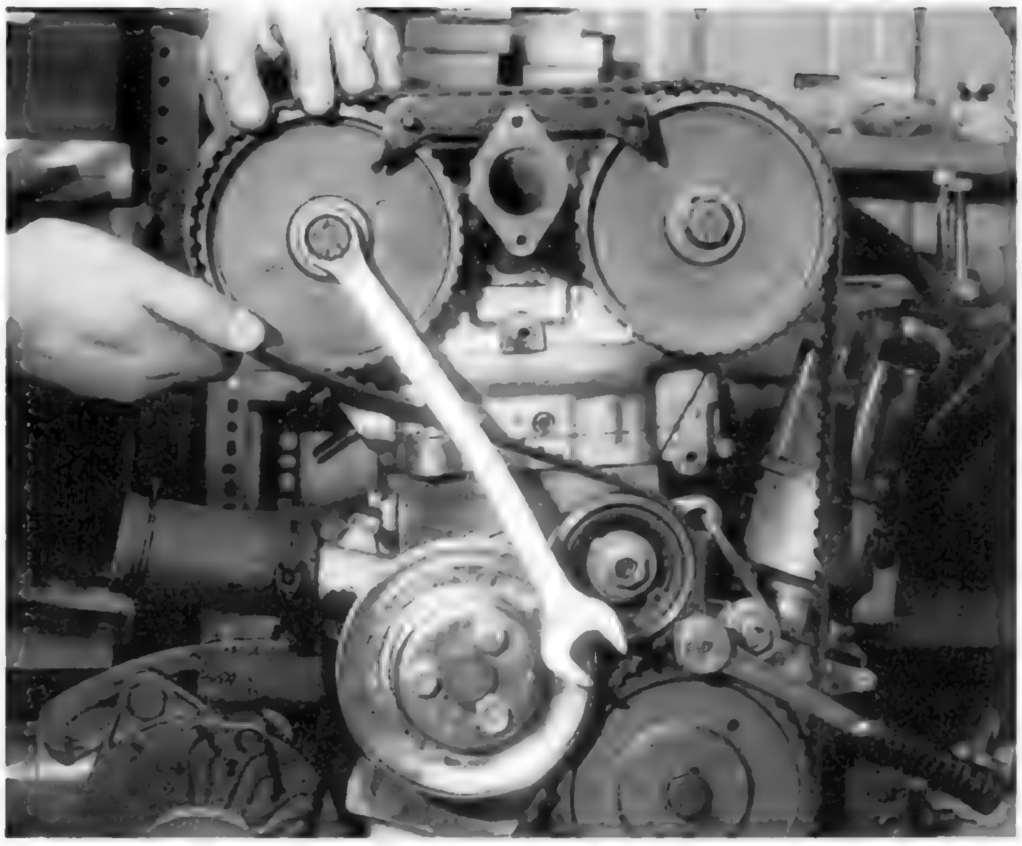
Before fitting the belt...
Bolt-type adjustable cam wheels should be loosened (the outer ring – not the cam pulley retaining bolt) prior to fitting the belt. Because the outer ring is free to turn, the applied belt tension will be equalized along the full length of the belt, so it is not necessary to carry out the tensioning procedure twice. The crank can be swung carefully to-and-fro to check they are free.
Note: Make sure the pulley bolts are secured before turning the engine more than a few degrees either way or valve damage will occur!



14/36: FITTING CAM BELT
Lock tensioner back and arrange belt to fit snugly around crank pulley and auxiliary driveshaft pulley. Note crank is at TDC on Nos 1 and 4, auxiliary driveshaft pulley timing mark at 34° from vertical. If pulley has been swapped, make sure it is timed up appropriately. Auxiliary driveshaft pulley is usually exhaust cam type – note alignment of timing hole relative to dowel/bolt line. Inlet type pulley will need timing hole on dotted line to keep shaft/dowel position correct. If fuel pump lobe has been removed/plugged, timing of auxiliary driveshaft is not needed.



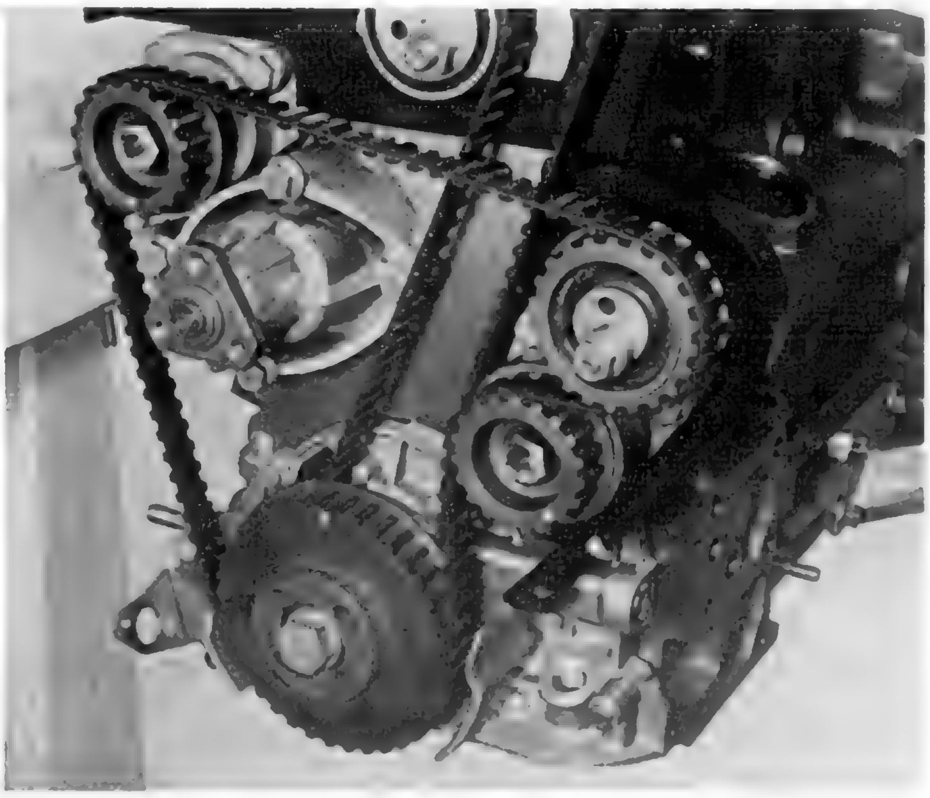
14/37: Left hand (Rabbit logo!) keeps belt in place on auxiliary driveshaft, right hand feeds belt over inlet pulley and tensioner. Slight repositioning of inlet cam wheel may be needed – hence spanner. With standard cams try to get pulley within ±1mm of timing mark. Idea is to remove slack from right-hand side of belt without turning crank.



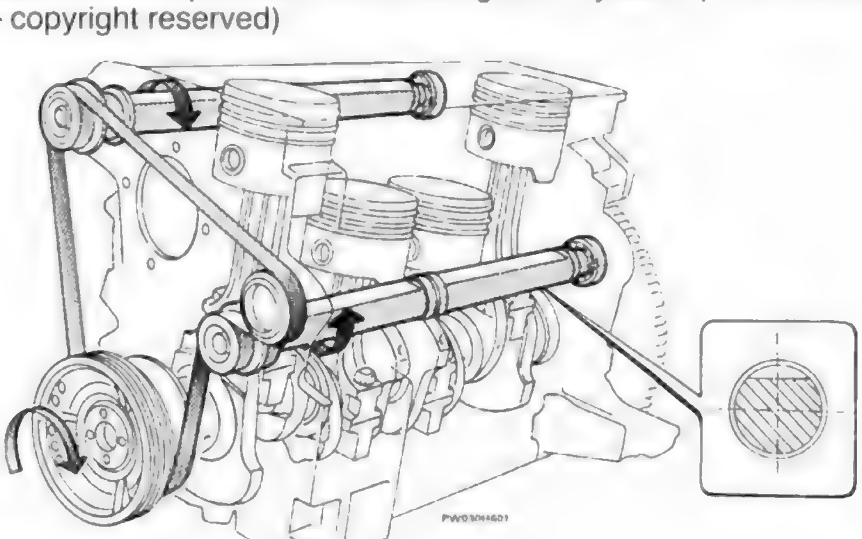
14/38: Feed belt over exhaust pulley – slight repositioning may be needed to remove slack between pulleys. Next stage is to apply tension to belt.

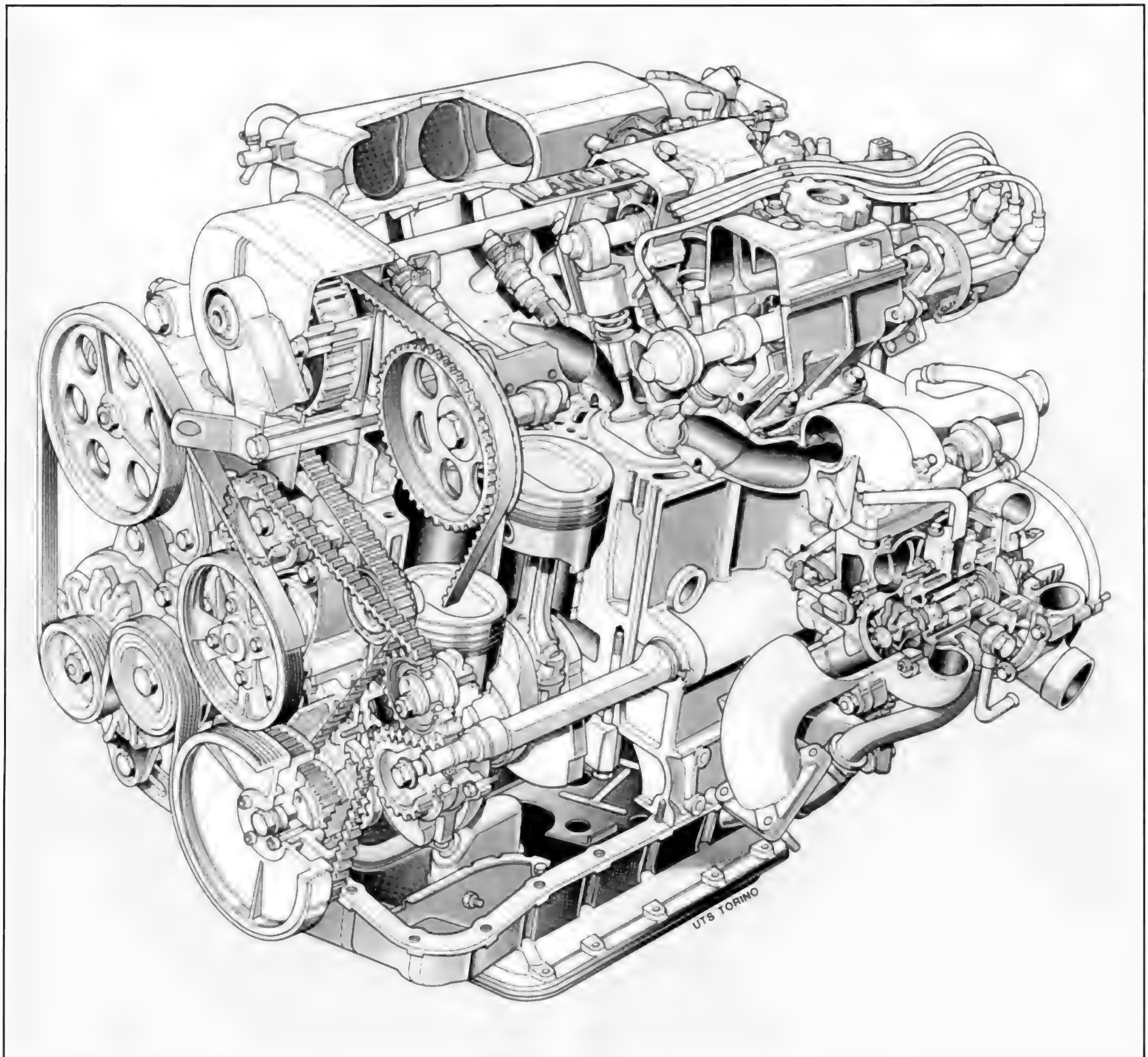
14/39: Apply tension to belt. Free tensioner and gently prise tensioner across with screwdriver an extra 1mm or so. This helps especially with 1" belt to ensure belt is really snug. Lock up tensioner nut and baseplate bolt. Rotate engine one revolution (clockwise viewed from crank nose) and retension to be sure there is no slack. Finally turn over engine at least two full revolutions to make sure nothing locks up. Do this by hand with plugs out, not with starter. If anything feels solid, it will be either auxiliary driveshaft (not correctly timed), valves touching either each other or pistons, or piston touching head. If this happens, turn engine back to TDC and check. If it is not the auxiliary driveshaft, head will have to come off. On transverse engines, if studs on left-hand side of block (viewed from crank nose) which secure driveshaft bearing housing are too long they can foul No 1 con-rod; similarly 124 1800 alternator bracket studs, so if they've been replaced and engine locks solid during this procedure, check to make sure they are not protruding into crankcase.

When belt tension is correct, it should deflect 8–9mm between cam wheels under a load of 22lb. As a rough guide, it should not be possible to twist belt through more than about 45–50° with forefinger and thumb between cam wheels.)



14/40, 14/41: Extract from Fiat 16v turbo manual shows correct positioning of balance shafts. Shafts add considerable weight to late block, but reduces engine vibration, thus helping to protect electronic components of fuel and ignition system. (Fiat Auto SpA – copyright reserved)





14/42: Cutaway shows 8v Integrale engine. When dismantling, note order of fitment of belt covers etc or reassembly will be a nightmare! (Fiat Auto SpA – copyright reserved)

VIBRATION DAMPING SYSTEM WITH COUNTER SHAFTS
(Reference drawing on previous page)

In addition to forces on the piston crowns, caused by expanding gases, the following are present in internal combustion engines:

- centrifugal intertial forces, resulting from the rotating masses;
- 1st and 2nd order alternating inertial forces, resulting from the masses with alternating motion.

The purpose of balancing the engine is to eliminate the vibrations caused by these imbalances during operation.

The imbalances caused by centrifugal forces and 1st order alternating inertial forces are eliminated by suitably counterweighting the crankshaft.

The imbalance caused by 2nd order alternating inertial forces is not usually eliminated in 4-in-line engines; it is left to the engine bearings to partially absorb it.

This engine instead adopts a system which cancels the vibrations caused by these forces; it comprises 2 counter-rotating shafts, with eccentric weights, located in the cylinder block.

The counter shafts are driven by a special double-sided toothed belt and set of sprockets which enable a speed double that of the crankshaft, and perfect synchrony with the latter, to be obtained.

Cam pulley alignment

If standard pulley teeth do not line up with the cam belt (within 1mm or so) it is worth removing them and redrilling the dowel hole. Minor misalignment can be cured by repositioning the cams; it is safest to turn the inlet anticlockwise and the exhaust clockwise because this withdraws the valves away from the pistons.

To redrill the pulleys, loosen the cam pulley retaining bolts, being careful not to allow the pulley to rotate and bend a valve (the crank can be turned away from TDC, but with big valves they may still clash).

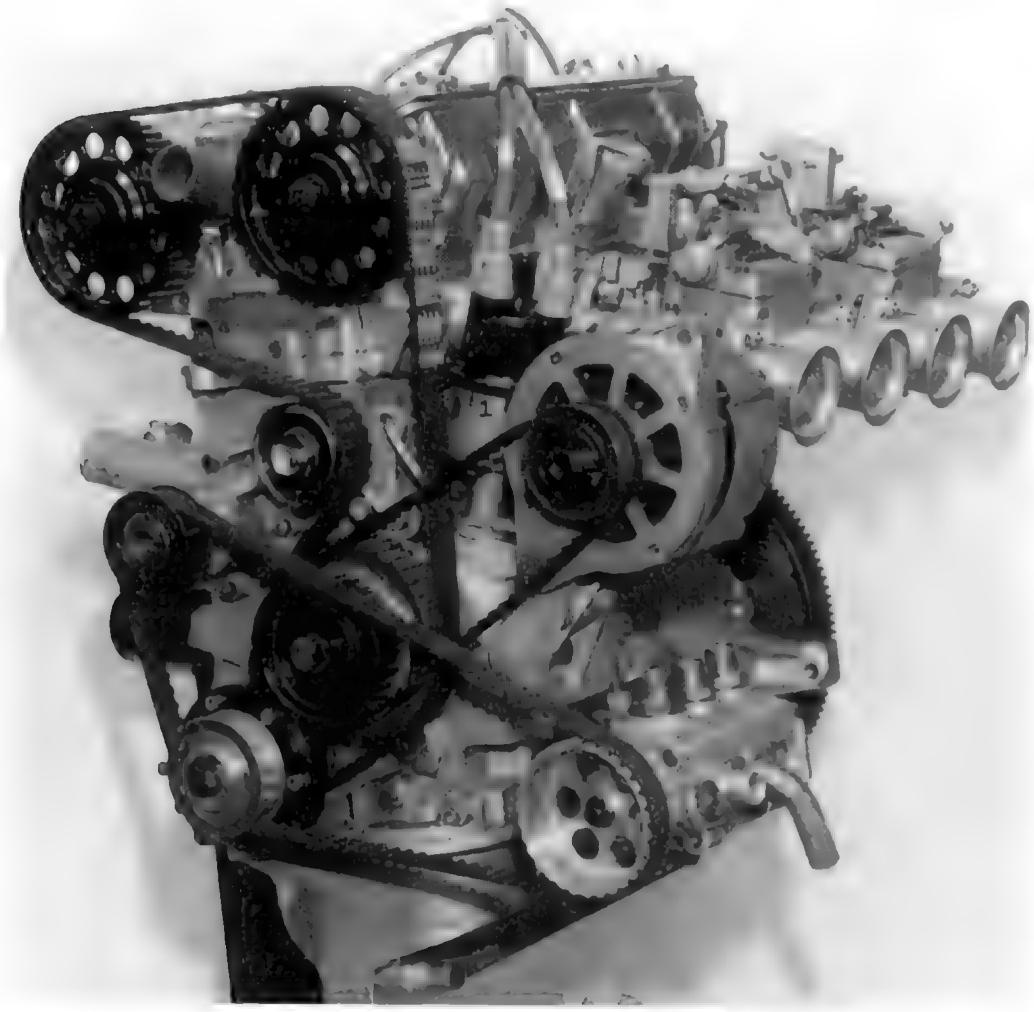
With the crank at TDC, slip the belt around the crank belt drive and inlet (or exhaust on late, reversed-port heads) pulleys, with the cam pulley off the dowel (the cam must be in its TDC position). Turn the cam pulley until the belt teeth line up and tap the pulley sharply with a hammer. The dowel will leave a new register mark on the pulley recess which can then be drilled out (5mm). Carry out the same procedure between the inlet and exhaust pulleys. 'Blueing' the back of the pulley helps. With cast iron cams a new dowel hole can also be drilled into the cam. En 40B in its hardened state is too hard to drill.

With GC peg-type verniers, if no plug holes align, turn the pulley outer ring to expose new holes.

Sump

Provided the engine turns freely after the head has been fitted, the sump can go on (this is the last opportunity to check that main and big-end bearing caps are properly tightened!). Check that the sump flange is flat (if necessary straighten, using a vice or hammer). Inspect the alignment of all the sump bolt holes and the condition of the threads in the block. It is well worth taking time to clean the threads (M6) because once the sump has been almost bolted up it can be very frustrating to find that the last bolt won't go in!

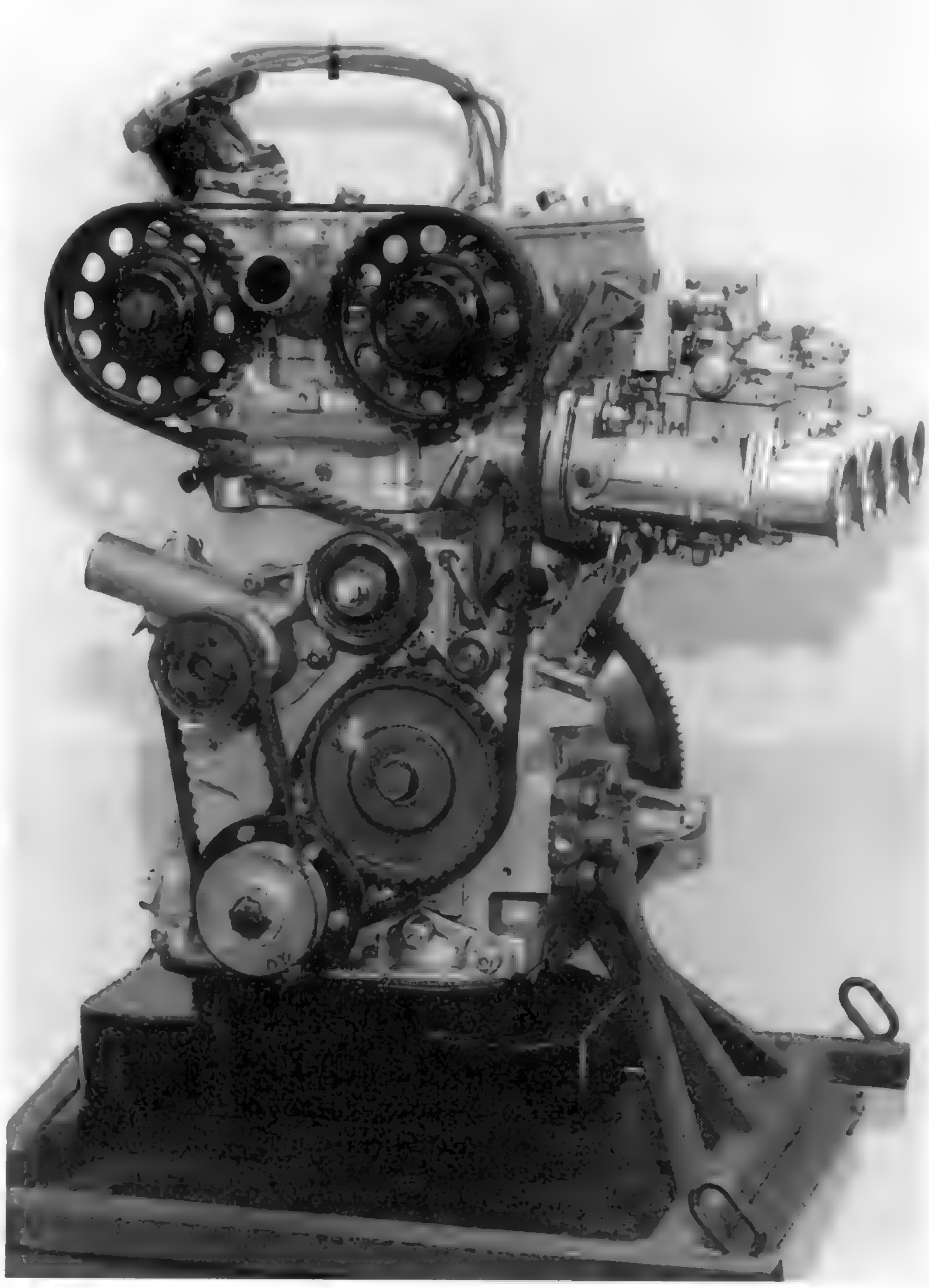
Place a small amount of silicon gasket on the block joints between the crank front and rear seal housings. Offer up the new gasket and bolt the sump in place. If a race sump has been fitted, use only four bolts initially and check crank rotation – minor clearance problems can be cured with a ballpein hammer. Bolt up the sump using new OE sump washers. It is a very good idea to Loctite them in place, especially on competition units. Don't over-tighten the sump bolts – it will only distort the gasket.



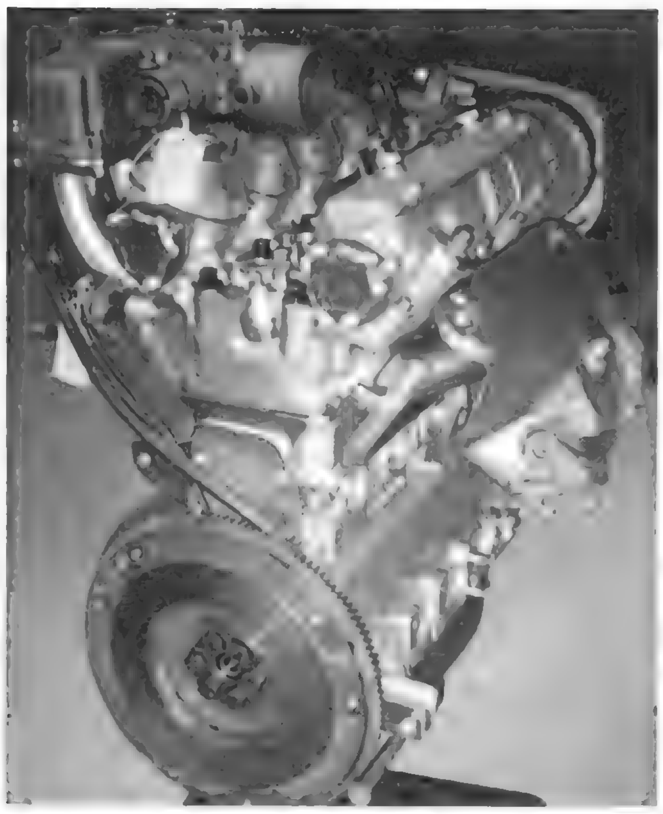
14/43: 2l Lancia, dry-sumped (with alternator) built for circuit racing. One scavenge pipe on Titan pump is missing and hoses have not yet been connected. Alternator fitting was extremely tight, even with Delta bracket. Alternator top bracket (not shown) bolts to cam box behind pulleys. Note use of Fiat early coolant outlet elbow. Water pump pulley is Morse Taper-Lock type; perfectly satisfactory if a bit on heavy side. Full spec comprised:

- 46/40 valves, alloy bronze inlet seats, nickel chrome exhaust bronze guides, head ported/blueprinted seats
- triple interference springs, GC lightweight verniers, 1" belt
- Stage IV cams
- forged pistons 11:1 CR
- fully prepared crank including stress relief/Tufftride
- dowelled steel flywheel, 7¼" single-plate clutch
- 48 DHLA (42 chokes), GC Beta manifold
- lightened, polished, shot-peened, balanced rods
- NGK B9 EGV plugs

Although engine never ran, and was later rebuilt to specification opposite, power output would have easily been 205bhp-plus @ 7500rpm.



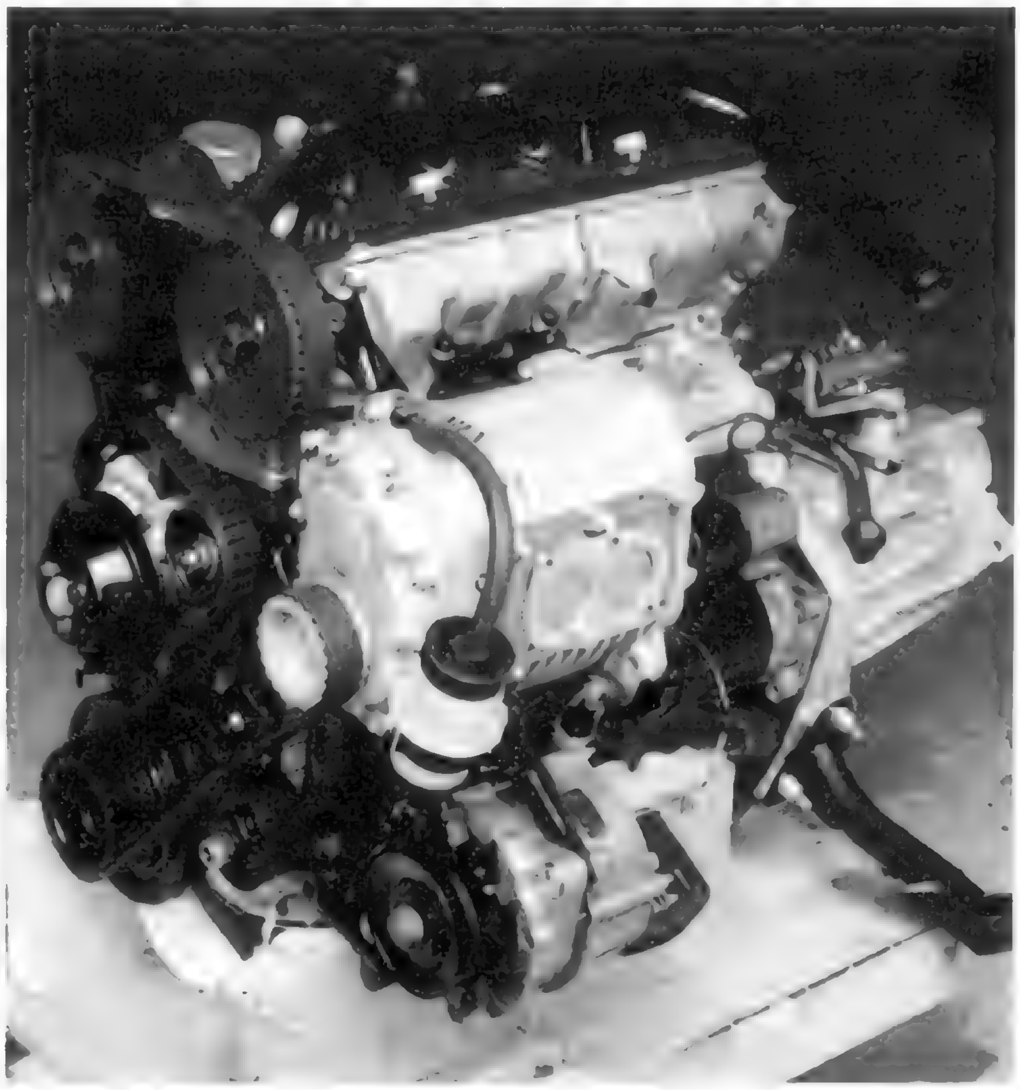
14/44: Lancia engine (preceding photo) was converted to rear-wheel-drive spec above. Note different sump, alternator removed, new distributor location with straight-shot inlet manifold, plus oil take-off plate for remote filter set-up. Engine went to Spain for 'veteran' circuit race in 124 Sport. A bit on the powerful side for a 124 gearbox, really.



14/45: Rebuilt 2l fuel-injected engine for Tony Dark's Spider. Injection is Bosch L Jetronic, where injectors are pulse actuated electronically; airflow is measured by flap valve in metering unit (not shown). Potentiometer on rear of throttle plate feeds positional data to ECU. How effective? Very tractable and fuel-efficient but not quite the torque potential of a 'one intake tract per cylinder' set-up due to wave interference in plenum. Exhaust-driven distributor not shown in this shot.

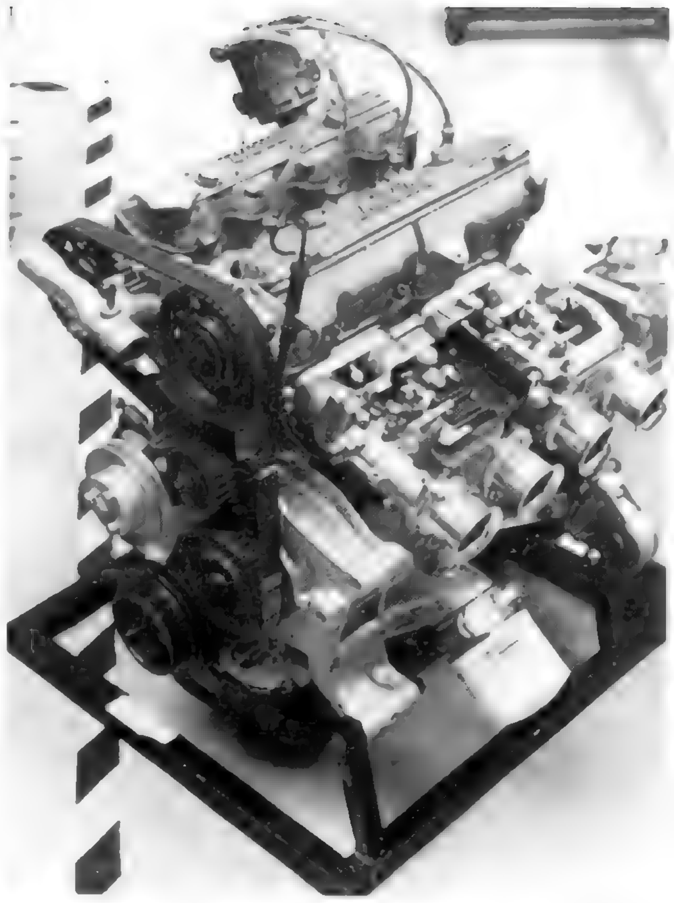
14/46: Leif Bengttson built this interesting 2l race engine. Dry-sump pump incorporates integral filter. Auxiliary driveshaft removed and direct drive belt fitted. A slipper pad will be needed near tensioner pulley, or repositioning of tensioner to prevent belt teeth touching when belt flexes; IIID cams, 44/38 valves, 11.5:1 CR, 48 carbs, estimated 190bhp. Electromotive fuel injection was being planned in late '95.

14/47: Michael Hockley, CNC machinist, hydraulics specialist and all-round engineering genius, who carried out much of the special machining and fabrication for Guy Croft Tuning in the years 1989–96.

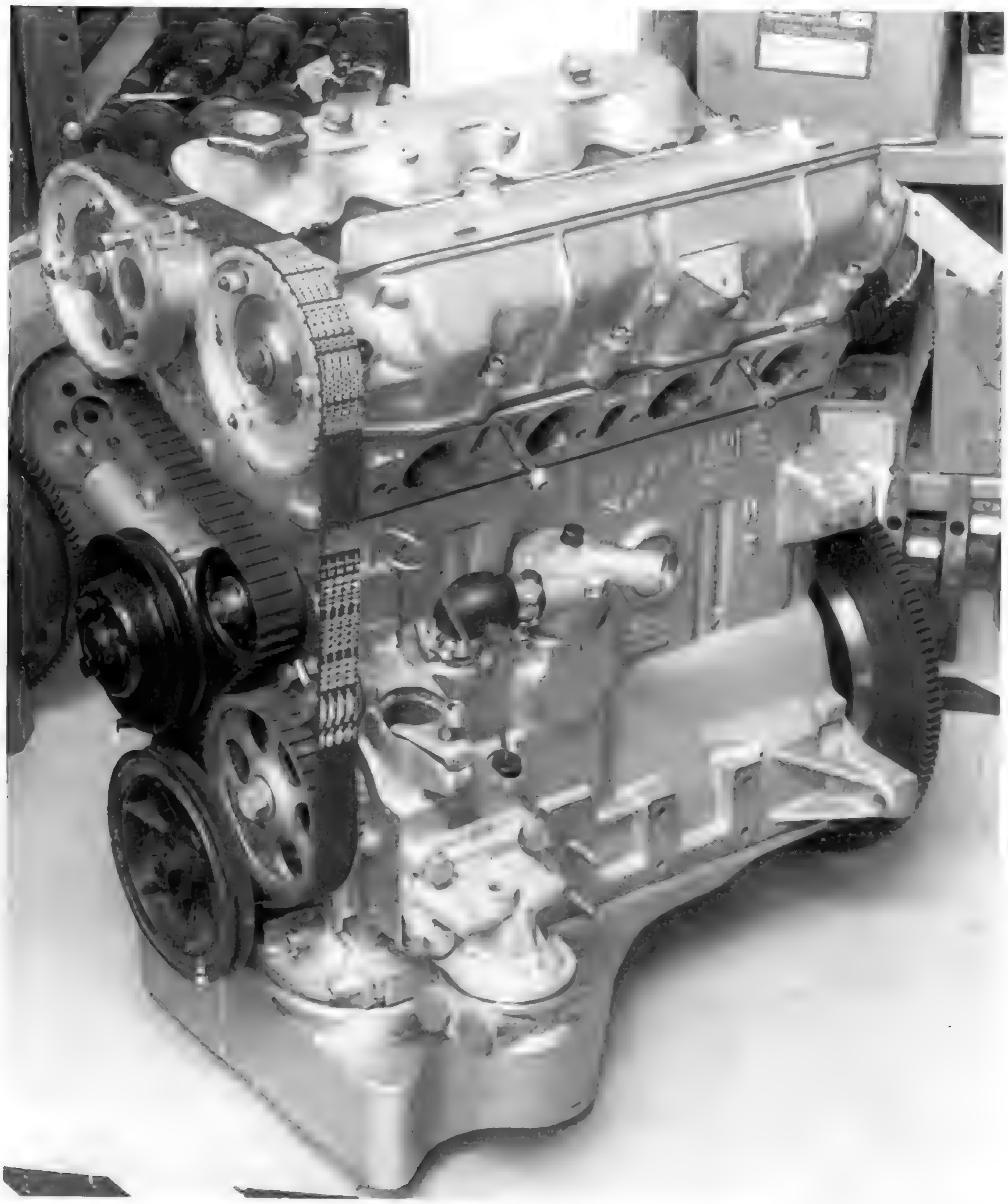


14/48: Dr Bob Sharp built up this Volumex unit for his Hawkridge Stratos replica incorporating GC-prepared head and block.

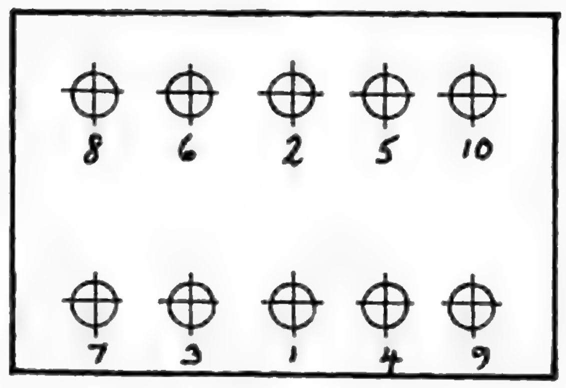
14/49: Graham Earl's engine for 124 CSA: 1800 85mm ⌀ pistons, Columbo St III rally cams. Note early GC verniers, changed for later, drilled lighter types. Distributor (not shown) is 124 CSA exhaust-driven type. Car was later sold to Japan and featured in Japanese magazine Tipo.



14/50: St III Guy Croft Monte Carlo engine with 10:1 CR, St II cams, 45s. Output was 172bhp @ 7200rpm. Late-pattern GC verniers and 1" belt fitted. With twin carbs fitted, alternator will be fitted to bracket upper mounting and braced to lower one (attached to engine frame) with strip of 5mm thick mild steel. Distributor drive has been swapped for Fiat 124 Sport BC type and cambox oil drain modified accordingly. Linkage is TLK/ID by Datum Carburettors, with 45 Dellorto DHLA.



14/51: Race engine for Morgan 4/4 owned by Jonathan Douglas features 86mm bore pistons (2.1l), 44/36 valves (unleaded seats for high-octane unleaded fuel), St III rally cams and early scalloped big-wing sump. Designed for use with single Ford V6 Essex carburettor, power in region of 165bhp.



14/52: Head bolt sequence – all TCs.

FASTENER TORQUE SETTINGS

FASTENER	ENGINE TYPE	TORQUE SETTING	NOTES
Con-rod bolts (nut and bolt types)	1586, 1592, 1608, 1756	38lbf ft	pre-oil threads
	1995	55lbf ft	
(bolt only types)	1995	25Nm + 50°	"
Main bearing cap bolts	All 1585, 1592, 1608 1756 and 2*l* (not 16v)	10mm dia bolt – 58lbf ft 12mm dia bolt – 83lbf ft	lubricate threads with Copper Ease and oil heads
	16v	centre cap bolts – 20Nm + 130° other cap bolts – 20Nm + 90°	"
Camshaft pulley retaining bolts (inc aux d/s)	all models	87lbf ft	use Copper Ease only, do not omit Fiat spring washer
Flywheel bolts	all 10mm dia	62lb ft	insert dry
	all 12mm dia	109lbf ft	"
Cam belt tensioner	17mm (AF) nut and late socket cap head type	32lbf ft	ensure new spring washer is used, or use Loctite
Crank front pulley nut/bolt	all early with nut	Fiat manual states 181lbf ft but 120lbf ft is enough	use Copper Ease
	all late with reverse-thread bolt	141lbf ft but 120 is adequate	"
Head bolts			
A	all early 10mm (not waisted shank hex bolts)	62lbf ft in 3 stages COLD (20–40–62)	as main bearing bolts
B	GC 12.9-grade race bolts	70lbf ft in 3 stages (20–50–70)	"
C	Fiat early stretch bolts (hex head-waisted shank)	15lbf ft – 30lbf ft – 90° – 90°	pre-oil threads and heads and drain 30min before fitting
D	Fiat late stretch bolts (spline type drive-waisted shank)	"	"
All 8mm bolts (8-grade or better) (including cap heads, black or s/s)	all TC	16lbf ft (22lbf ft on clutch cover with 12.9-grade bolts)	lubricate threads with Copper Ease. Make sure no beadblast grit in threads (esp head)

Retorque – head bolts

Notes:

1 Bolts used with early (pre-Astadur + 130 TC type) gasket should be type A or B and must be retorqued after 300 miles. This must be done with the head COLD. Slacken the bolts in turn (¼ turn) and retorque.

2 Where an Astadur gasket (or later Fiat equivalent polymer type) is used, bolts must be retorqued if type A; others do not require retorque.

3 All bolts can be safely used four times. If in doubt, replace.

Conversion table:

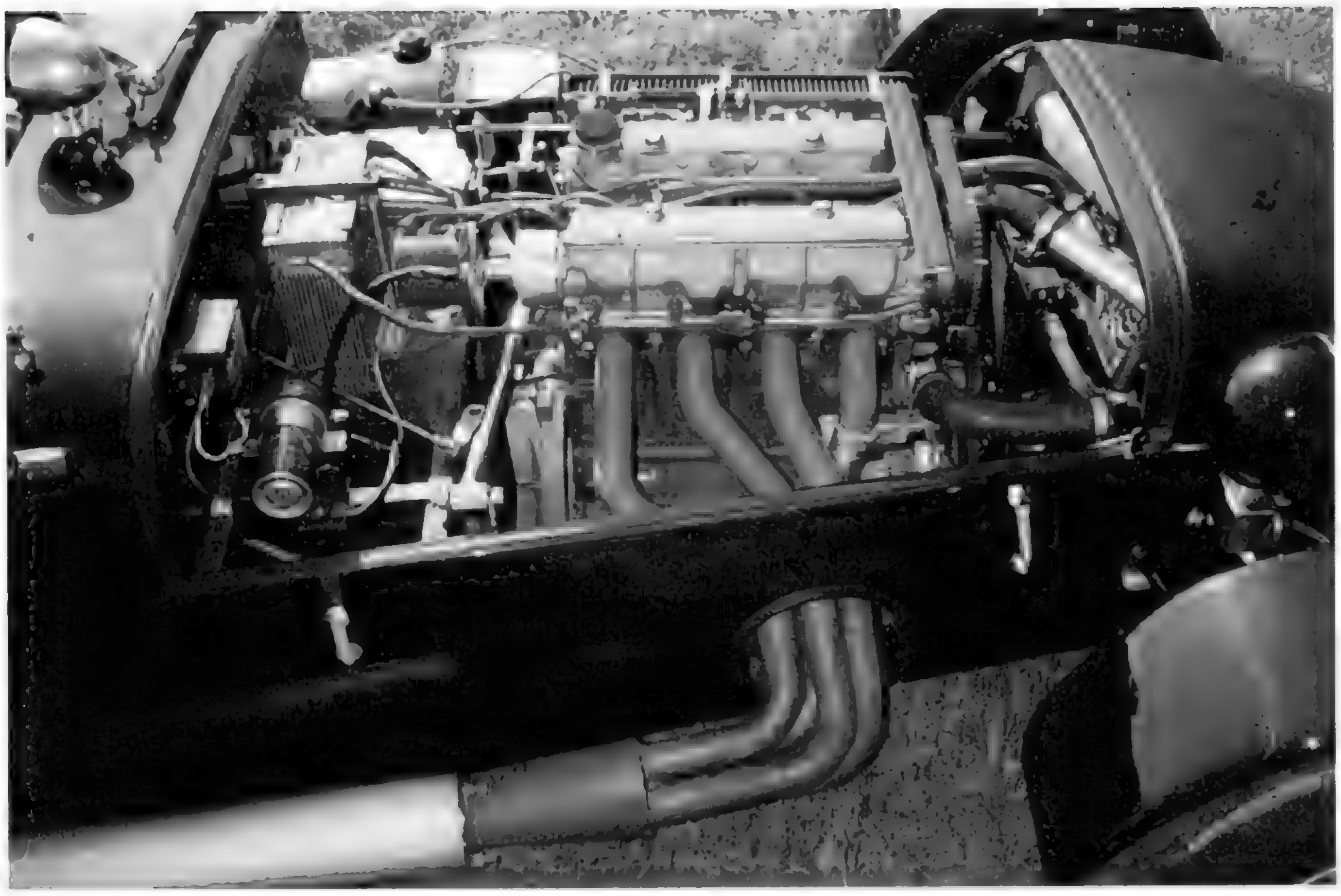
To convert from Nm to lbf ft	multiply by 0.74	(10Nm = 7.4lbf ft)
To convert from lbf ft to Nm	multiply by 1.4	(10lbf ft = 14Nm)
To convert from lbf ft to kg m	multiply by 0.14	(10lbf ft = 0.14kg m)
To convert from kg m to Nm	multiply by 9.8	(10kg m = 98Nm)
To convert from kg m to lbf ft	multiply by 7.2	(10kg m = 72lbf ft)
To convert from da Nm to Nm	multiply by 10	(10 da Nm = 1Nm)

CHAPTER 15

EXHAUST SYSTEMS

The exhaust system is comprised of the manifold, tailpipe and silencers. Its primary function, of course, is to carry away waste gases from the combustion cycle safely and reduce noise levels, but the system can be designed in such a way as to enhance the power output from the engine.

The exhaust valve opens before Bottom Dead Centre and is still under considerable pressure from the firing cycle. At the same time, a pressure wave generated by the sound energy from the explosion in the cylinder travels out of the exhaust port and into the manifold. As the piston passes BDC and starts to move up the bore, the piston itself imparts further energy to the process. By suitable tuning of the system, the sound wave can be used to improve the evacuation of waste gas from the cylinder. Most people are aware of the common configurations of 4-2-1 and 4-1 associated with exhaust manifolds, but fewer have the knowledge of how these differing set-ups work, or of how they may be complemented by the design of the system as a whole.

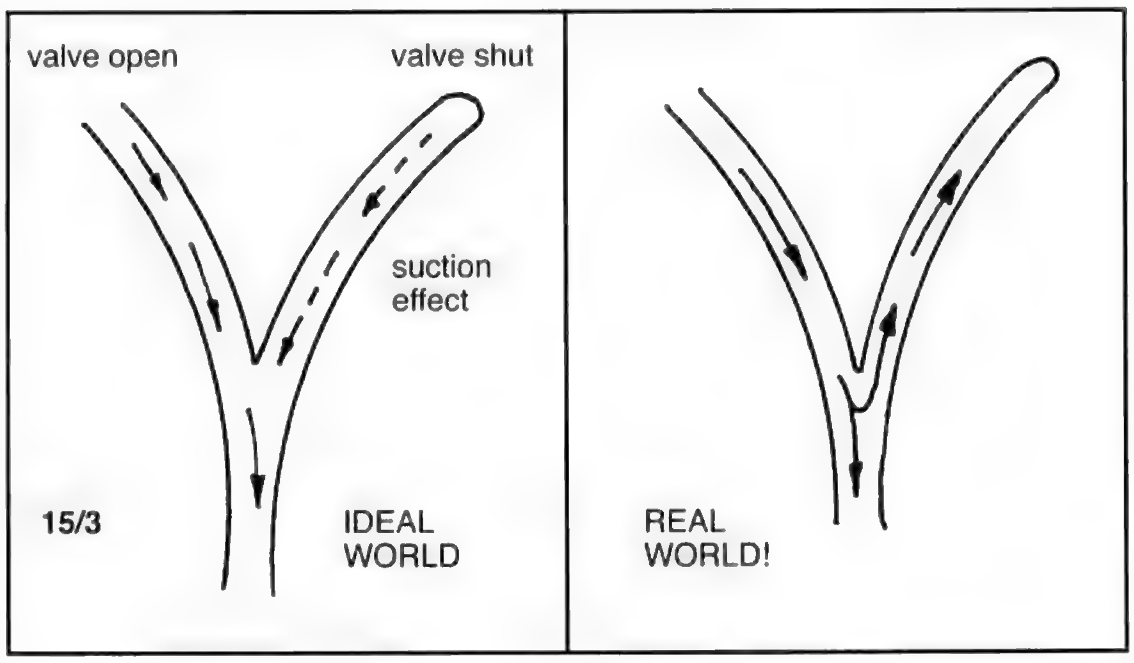


15/1: 4-1 manifold and single silencer on Bentec 7 designed and built by talented Swedish engineer Leif Bengtsson. Engine is ex-Ritmo (Strada) 125 TC with Beta 1600 pistons (approx 10.5:1 CR), Weber 45DCOEs (36 choke). Note special carrier for 131 S147 distributor and connection of water outlet at head rear to main elbow at front to ensure circulation at back of head. Gearbox is 5-speed Ford on special adaptor.

15/2: Claes Eklund owns this nicely prepared Bentec 7. Engine is standard 132 GLS 1592cc with production manifold and twin downpipes forming a 4-2-1 set-up. This layout works well with standard cams, of relatively short duration and low lift at TDC. Engine retains external thermostat: ensures quick warm-up but too hot for any engine with more than about 30% extra power.

The normally aspirated (*ie* not supercharged or turbocharged) engine relies entirely on the pressure differential between the cylinder (in a state of vacuum on the intake stroke) and the inlet tract (which, beyond the throttle plate, is at atmospheric pressure) to fill the cylinder when the inlet valve opens. Therefore, any increase or extension of the vacuum condition around TDC generated by the outgoing exhaust gas can only lead to a power increase. On a 4-2-1 manifold, the primary pipes (primaries for short) are linked together at a certain distance, and similarly the secondaries, prior to merging into the tailpipe.

Why are the primaries joined? A common misconception is that the outgoing gases in one pipe will create a depression in the linked primary pipe much in the way as they do in the cylinder, but sadly this is not the case. Extensive experiments have now proved



beyond all doubt that the sound wave (travelling at 1200ft/sec-plus) exiting from the exhausting cylinder can 'U-turn' quite easily around the junction and into the adjacent primary. Clearly this is capable of preventing the linked cylinder from exhausting efficiently.

In fact this condition is far from being the disadvantage it first appears to be. The high-speed pressure wave, or pulse, if allowed to enter a primary pipe on which the exhaust valve is closed, is reflected off the closed valve, again U-turns around the junction and re-enters the cylinder as the piston is around BDC. The great advantage of this is that it leads to an extended vacuum condition in the cylinder around TDC. **(15/4)**

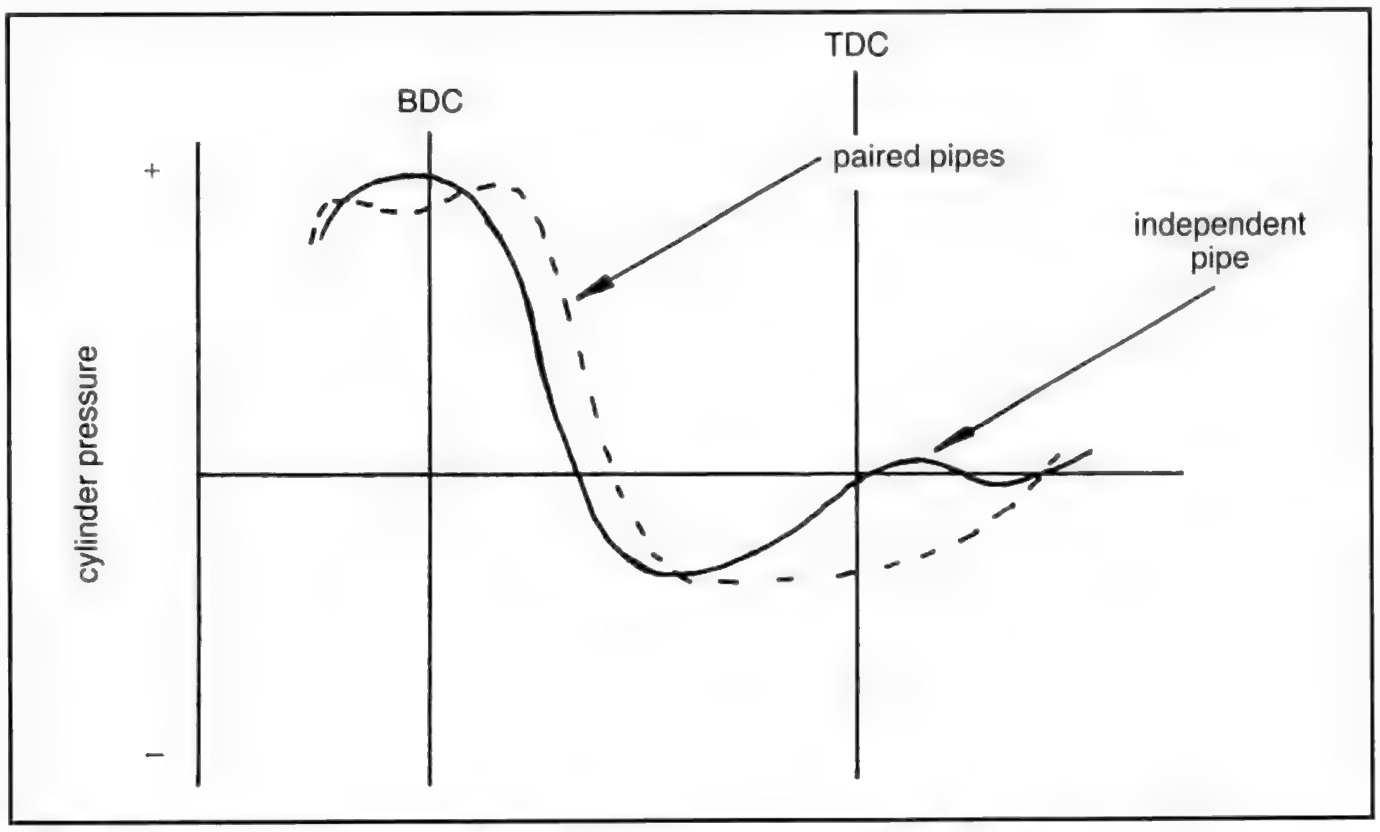
The extent of this phenonomen depends on a series of factors; if the exhaust gas speed imparted by the piston movement is added to the pressure wave because they happen to be moving in the same direction, the effect around TDC will be greater than if they are opposing. Principally this is governed by the engine speed and the lengths of the primaries (and hence the time for the pressure wave to travel the distance involved).

The paired cylinders must be chosen so that the valve on the linked cylinder is closed for the full phase of operation. On the Fiat/Lancia TC the firing order is 1-3-4-2 and since the engine requires 720° of crankshaft rotation to complete the full working cycle on all cylinders, the firing interval between cylinders is 180°. Therefore, the furthest apart two linked cylinders can be is 360°, *ie* 1 and 4 can be safely branched together as can 2 and 3. The linking of the primary pipes in this way is known as interference.

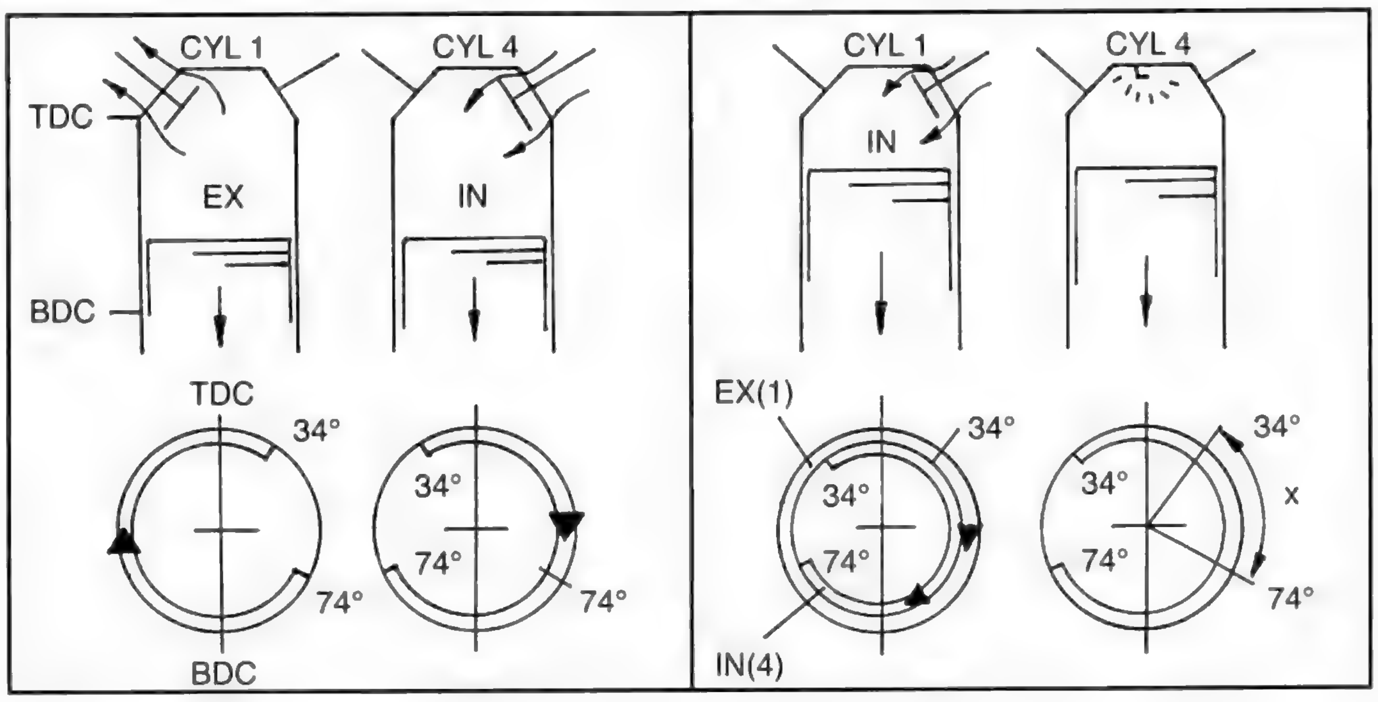
In diagram **15/5** it can be seen that No 1 exhaust valve opens at 74° before BDC (74/34 cam) when No 4 cylinder (360° behind) is in its intake stroke and No 4's inlet valve is 74° before BDC. When No 1's exhaust valve closes at 34° after TDC and No 1's inlet cycle has commenced, both valves are closed on No 4 for the firing stroke. Obviously, a short time after ignition, No 4 exhaust valve will open at 74° before BDC and the phase 'x' represents the period of time available for the primary pipes to be clear of exhaust gas pressure prior to this.

Clearly, back pressure from stalled gas trapped in the manifold, or residual wave action, can have an adverse effect at this stage and the shorter the 'x' interval (with longer-duration cams) the worse the problem. At a crank speed of 7500rpm, with 50/90 cams the 'x' interval is only 0.0009sec.

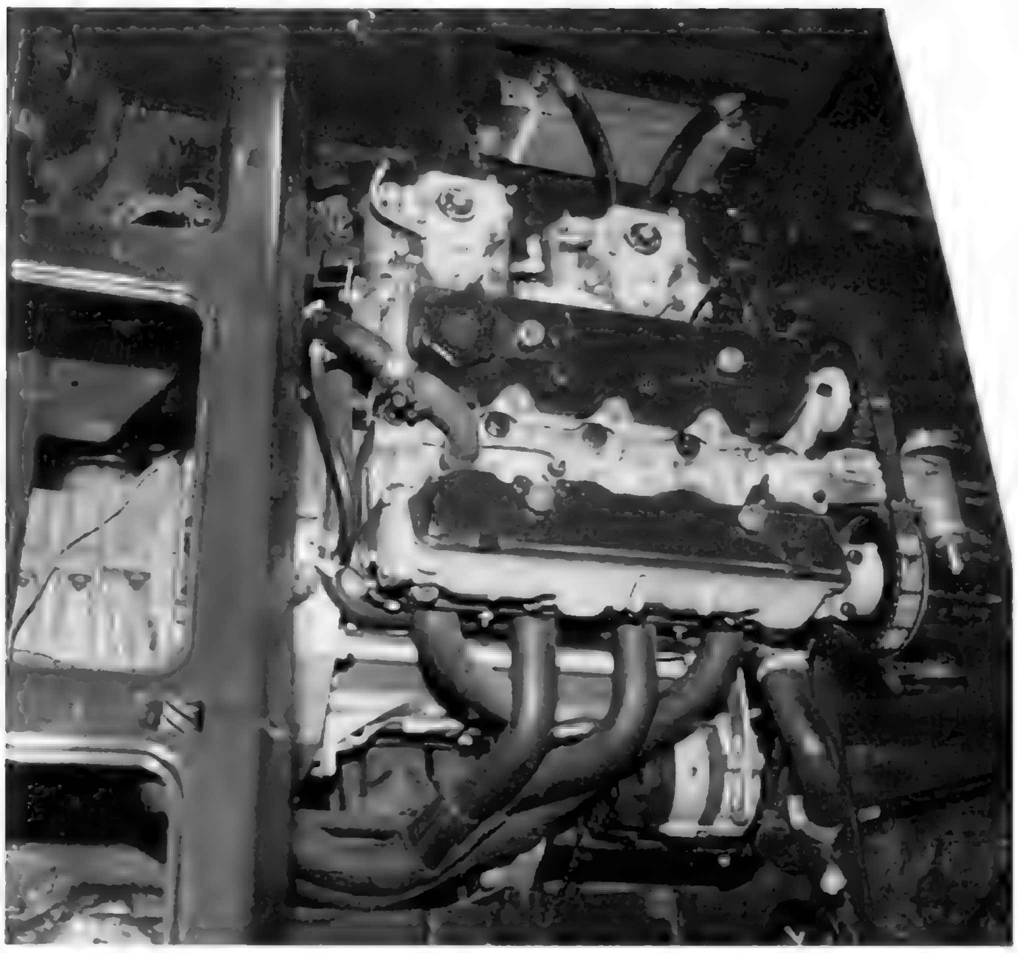
A number of factors govern the overall efficiency of the 4-2-1 manifold. If the pipe bores on the primaries are too large, the pressure wave will be diminished and the exhaust gas velocity will drop significantly. Too small and back pressure may result. The use of a deliberate mismatch on the port/manifold interface, where the manifold is larger than the port, inhibits the reverse-flow of exhaust gas back into the cylinder (not to be confused with the pressure wave, which travels freely



15/4: Showing extended low pressure in cylinder around TDC where exhausting cylinder linked (paired) to closed pipe (valve closed).

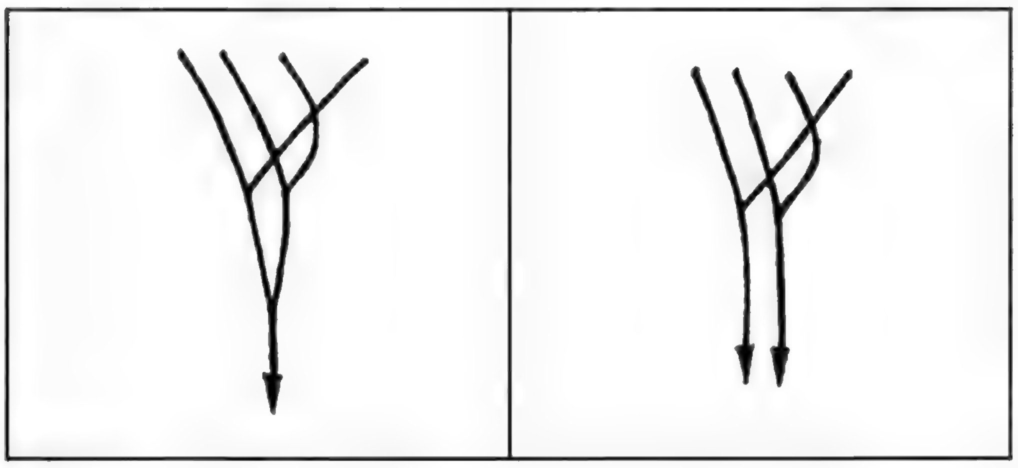


15/5: Valve phasing on paired cylinders 1-4.



15/6: Eric Cox prepared this 2l-engined Skoda for drag racing. 2l Fiat mid-mounted with Renault transaxle box, features GC ported head, 45s, 9.6:1 CR. 4-2-1 manifold primary pipes a bit on short side but OK with standard cams. Output around 155bhp @ 6300rpm, 142lbf ft @ 3800rpm. Massive duct (not visible) on nearside rear wing feeds cooler.

15/7: Paired and split secondary pipes utilizing interference principle in primaries.



through its medium, *ie* exhaust gas, in the same way as pressure waves travel through air – whether moving or still air).

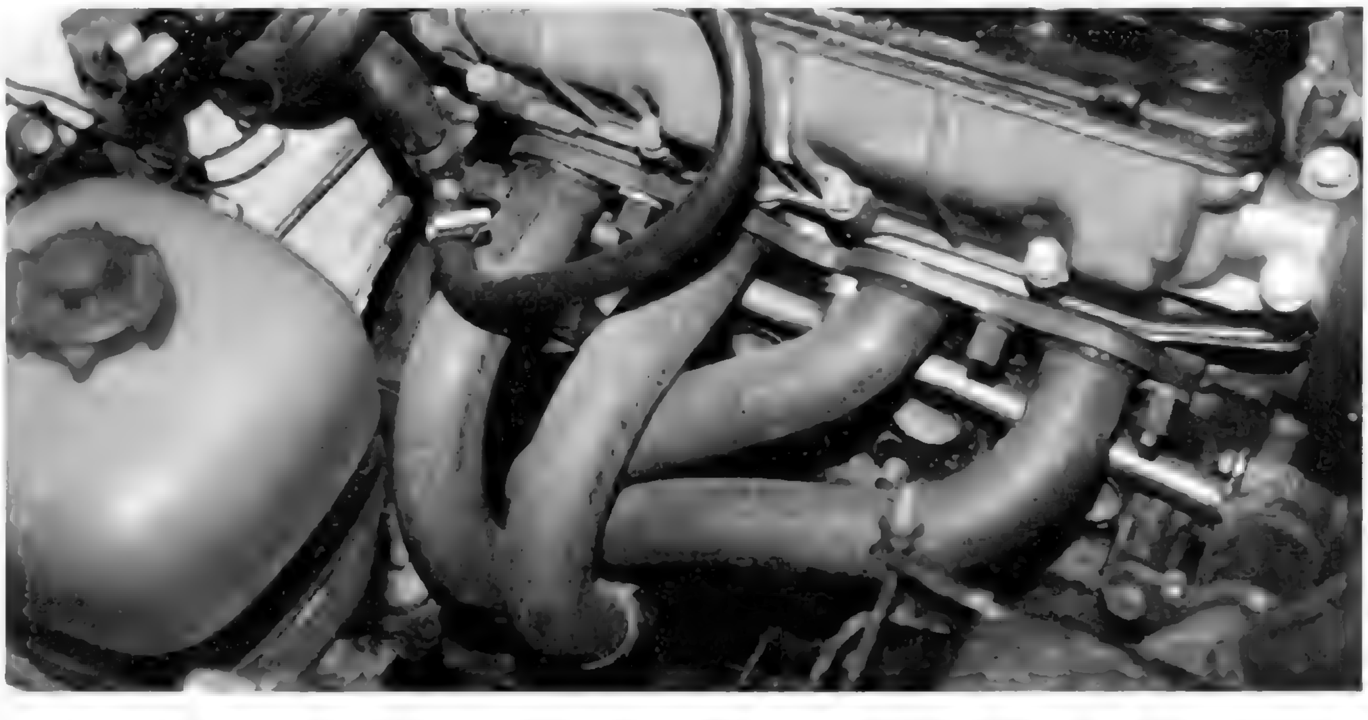
This anti-reversionary effect requires a step of +5–7mm on the diameter so that if the exhaust valve throat diameter is 33mm, the primary pipes should be 38mm (1.5"). However, it may safely be assumed that ±15% tolerance exists on this figure, based on good dyno results obtained from smaller and larger sizes. The secondary pipes should be between 25% and 35% larger to carry the extra volume of gas from two cylinders.

Because of the complex factors governing the behaviour of the exhaust gas, and despite numerous promising equations offered to calculate the pipe lengths, actual derivation of pipe lengths tends to be a case of 'suck it and see' with the best manifolds derived from dyno-testing. Part of the problem is the confused interaction between the pressure wave and the piston speed, and what may work well at one speed may be less suitable at another. Likewise, the camshaft direction and acceleration (principally governing the lift at TDC) are significant.

Obviously, the standard manifold tends to be the manufacturer's expedient answer to the relatively high cost of producing a nice 4-2-1 tubular steel set-up, as seen on the 130 TC. This cast manifold works very well with the mild standard cams, and moderately well with competition profiles up to around 300° duration and 3mm LATDC; beyond, and possibly including that, a properly designed tubular set-up is vital to maximize the torque output. As a general rule, the primary lengths should be at least 12" and preferably 17–19" long, and matched in length (to equalize pulse intervals). The short-stroke (71.5mm) 1585cc tolerates shorter pipes than the 2*l* because of its lower piston speed.

15/8: A 4-2-1 manifold on Ray Carden's 2l grasstrack car. Primary and secondary lengths are 17". This St II engine develops approx 175bhp and has proved highly competitive.

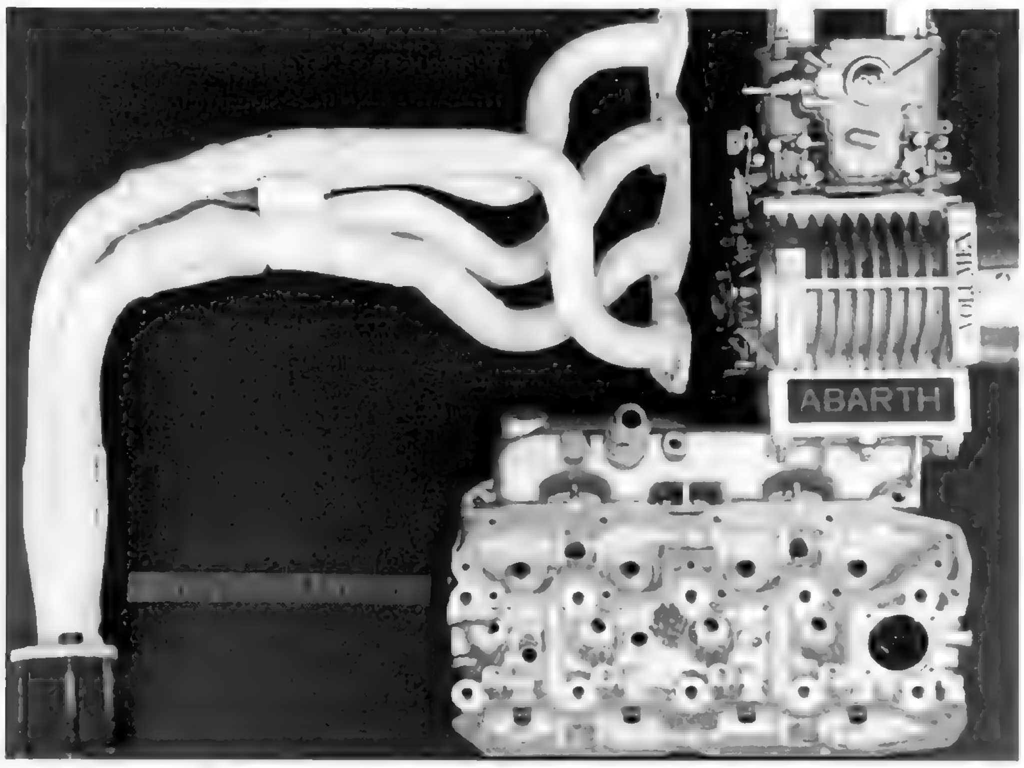
The primary purpose of the secondaries, of course, is to blend the primaries to the tailpipe for routing into the silencer, but keeping the secondary pipes separate helps to prevent the wave action in the primaries from contaminating the adjacent cylinders. The optimum lengths of the secondaries are the same as for the primaries, though a reduction in length of up to 25% should work well. Again, pipes should be of equal length. It is acceptable to use a manifold with split secondary pipes provided that the primary pipes are long enough; 14–16" is the realistic minimum, depending on stroke – too short and



15/9: Not an easy job but decent 4-2-1 manifold can be made to fit in LHD 124 Coupe. RHD is even more difficult due to position of steering box.

15/10: Attractive 2l Fiat-engined Westfield built by Martin Gallacher of Aberdeen. Slight variance of primary pipe lengths on nicely made stainless 4-2-1 seems not to inhibit power output of around 174bhp @ 7000rpm, with IIIA rally cams, GC ported 130 TC head, 45s, 10:1 CR. Digiplex proved unreliable for some reason (never established) and was later ditched for Bosch electronic (non-mapped) system. Bonnet clearance ¾", ground clearance 4¾". Quick hillclimb car.

15/11: GC engine No 168: big-bore (86mm) 1800 in Mick Woods' Gp 4 124 Abarth Spider. Engine features 46/40 valves, Abarth rally cams, forged pistons, 10:1 CR, shown with 44 IDF carbs. Manifold is original Abarth works version 4-2-1, 19" lengths; single large-bore tailbox. Early water outlet elbow runs direct to radiator. Large-bore hose along bulkhead carries breather fumes to catchtank. Exhaust manifold notoriously difficult on these models due to steering idler boxes.



15/12: This 4-2-1 Ansa Monte Carlo manifold accepts 175bhp on Tom McGaffigan's Volumex-blown Monte Carlo.

15/13: Author swapping main jets during a series of dyno runs on GC engine No 88, St II 2l Lancia for Nigel Peyton's 246 Replica. Engine still yielded 172bhp @ 7000rpm with 142lbf ft torque at 5500rpm despite relatively small (31mm) primary ID Ansa 4-2-1 (37mm exhaust valve). Engine is inclined at 20° to accommodate Beta sump pan. Large tower to right is thermostatically controlled heat exchanger. No thermostat is fitted in engine for these tests. (Superflow rig.)

chronic wave action in the primaries will upset the firing cycles at low speed.

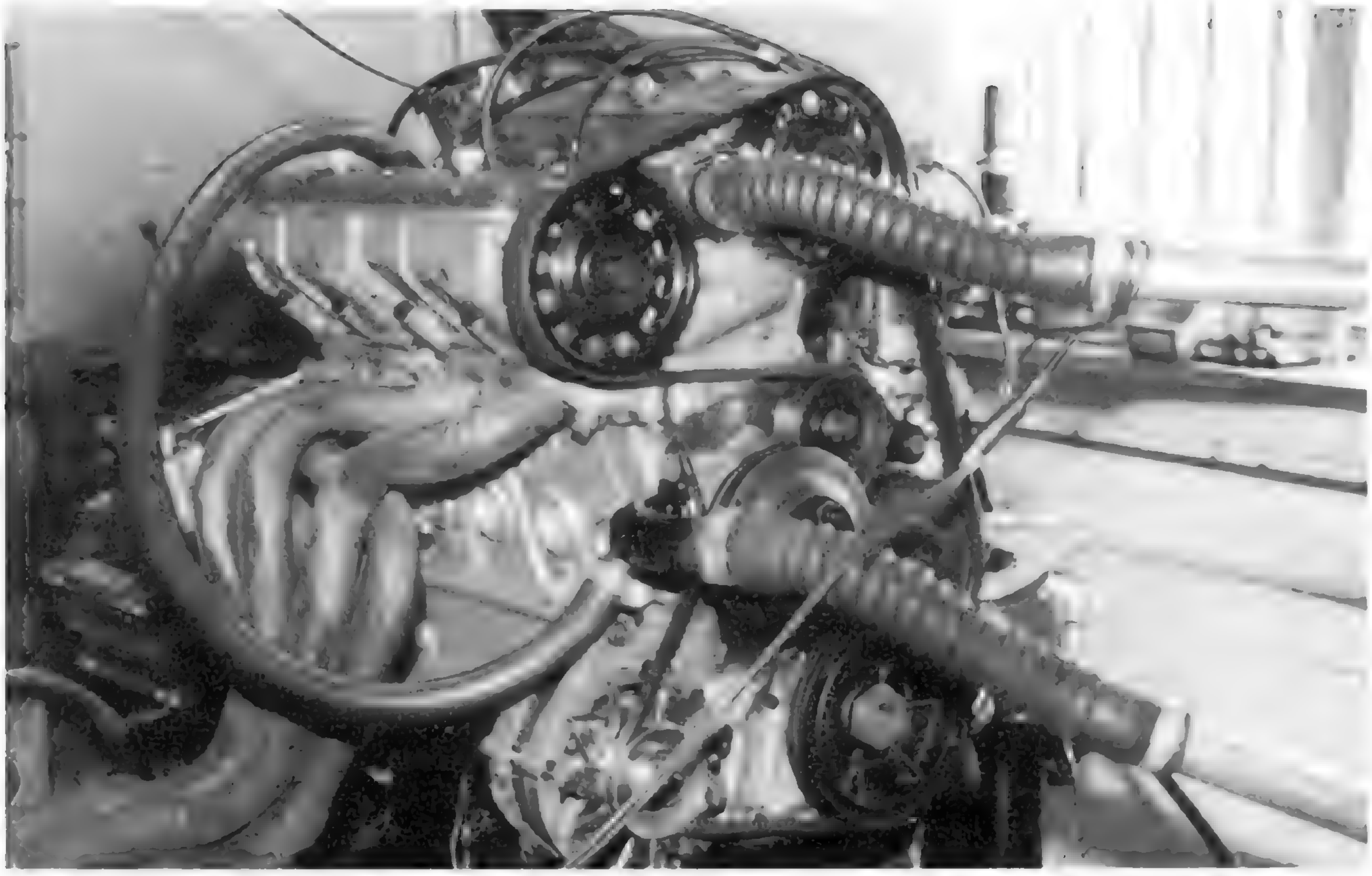
Tailpipe length is not crucial and depends mainly on the available space, indeed the decision to use a 4-2-1 itself is often made on this basis because the primary pipes on a 4-2-1 have the same effect, due to the interference principle, as a 4-1 with primaries of double the length.

4-1 manifold

A 4-1 manifold will tend to give better top-end power than a 4-2-1 when used with high-acceleration, long-duration cams. The primary pressure wave is of such magnitude and the 'x' interval so short, especially at high piston speeds, that it is better to keep it separate from the other branches for as long as possible. Circuit racing tends to have long straights requiring top-end horsepower for flat-out speed, and 4-1 systems are most commonly seen on race cars. Rally requires frequent use of mid-range for accelerating away from tight corners and gradients and, coupled with cams developing a broad spread of mid-range torque, 4-2-1 systems are often more effective.

The 'x' period mentioned earlier can be reduced to as little as 40° with 320° cams, and there is a serious risk of wave action persisting between linked cylinders. As with a 4-2-1 manifold, the lengths can easily be optimized on a dyno by 'cut and shut' on the primary pipes, but the absolute minimum length on a 2*l* is 22", 34–36" being better. Anything less than 22" can cause chronic interference between the cylinders and the engine may not 'rev-up', even *off-load*, until the crank speed reaches high revs. The diameter of the primary pipes should be the same as those used on a 4-2-1, with the secondary pipe (which is in effect the tailpipe) of 25% larger diameter.

Systems can be made of mild steel or stainless; 16 Gauge (0.064") is commonly used since it gives a good combination of strength, longevity and ease of welding, although 18 Gauge can be used on superlight layouts. Stainless steel cannot be



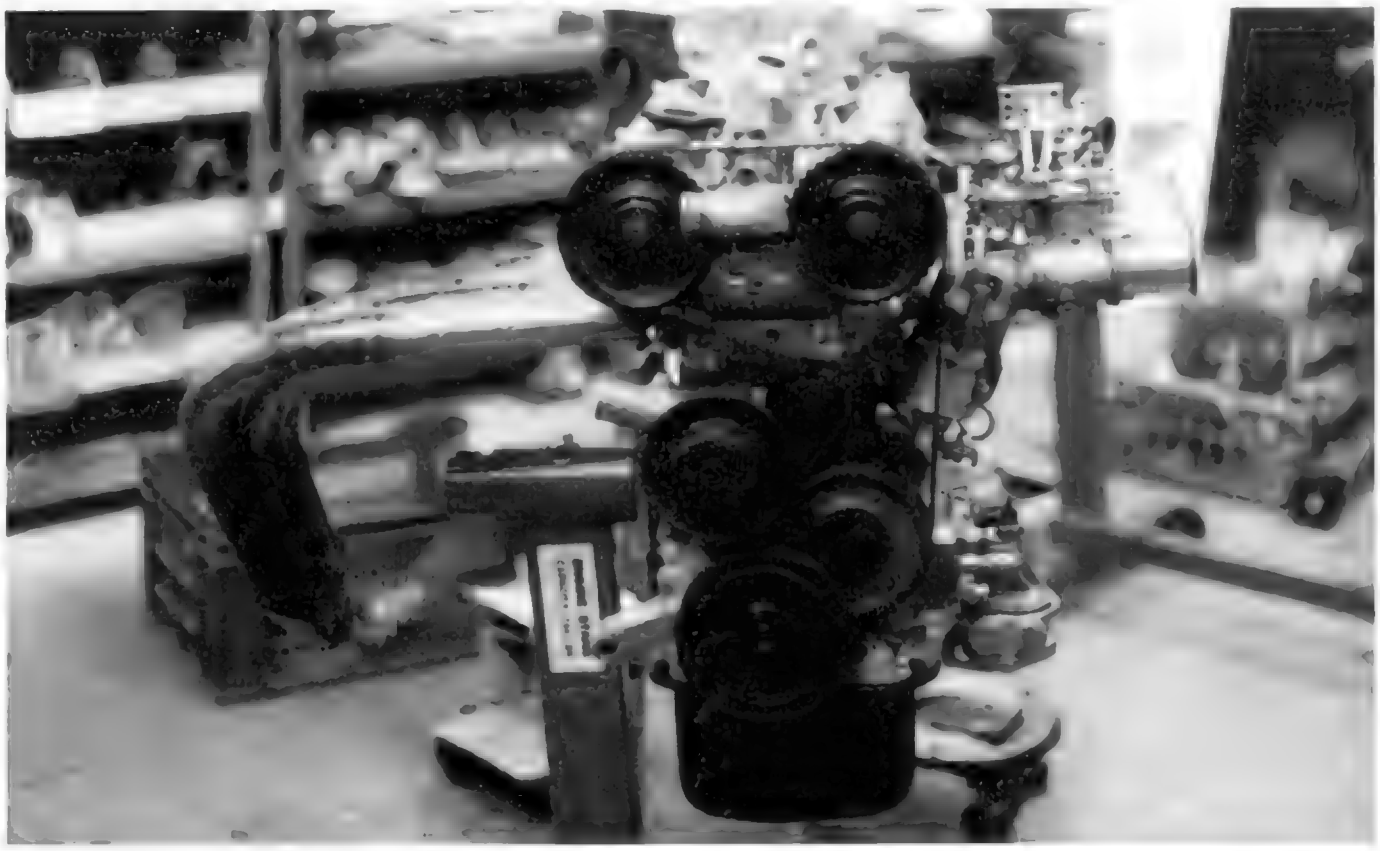
formed into particularly tight shapes, so mild steel is preferred and certainly in these installations the tube should be 'hot formed' to prevent stress cracking. 'Heat wrap' material should be avoided at all costs.

It is often stated that the reduction in heat transfer from the manifold leads to a power increase. Whether this is actually true is highly debatable, but it is certainly true that enclosing the pipework leads to rapid stress cracking, which is almost impossible to repair. If shielding must be used (for example when the manifold is routed near the carburettors) a stand-off of at least ½" should be allowed.

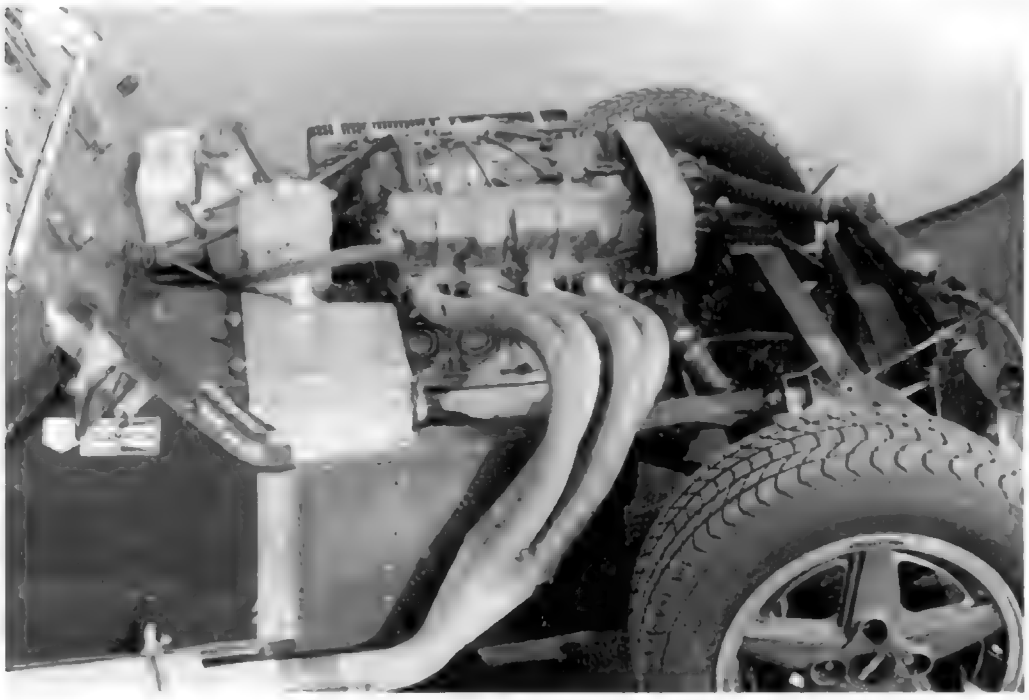
Silencers and back pressure

The purpose of the silencing arrangement is two-fold; principally its purpose is to reduce the noise level to an acceptable

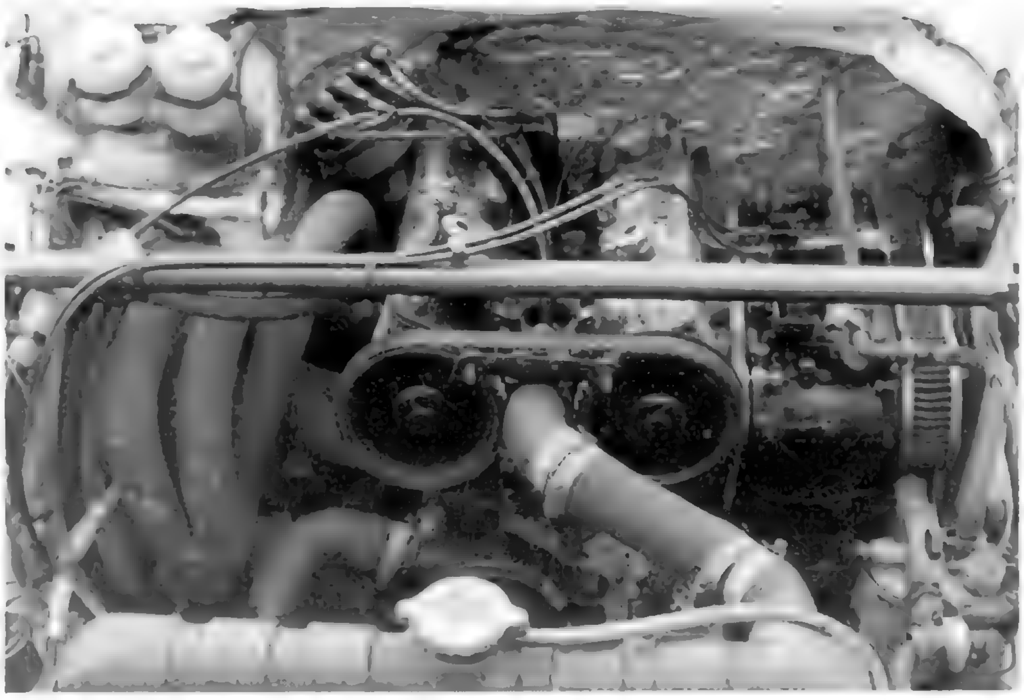
15/14: GC engine No 110, Dave Light's 246 replica power unit on 'Go power' dyno. Despite having rally cams, 4-1 manifold yielded good results. Beta engine was rigged with Fiat alternator for test. Note coolant hose from rear of head connected back into pump to keep rear end of head cool. Lack of oil heat exchanger on this rig ruled out extended full-power runs. Belt whip on alternator is about ±¾" at full speed! Use of notched belt is essential to prevent it flying off. Tappings in primaries are for exhaust gas analyzer.



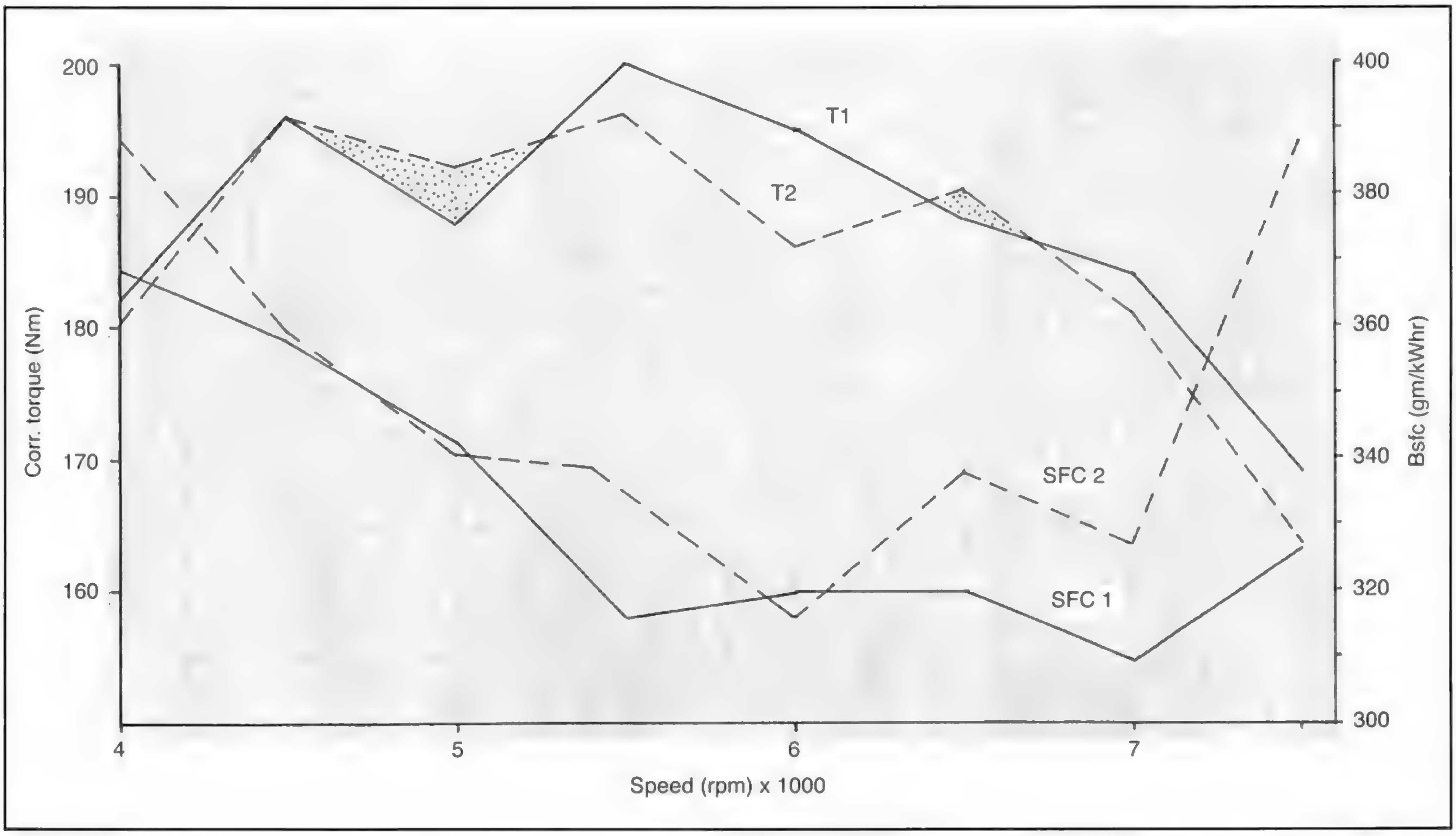
15/15: 3ft primaries on St III race 1800, Guy Croft engine No 57. Forged pistons, 10.7:1 CR, 48/67, 65/46, 11.3 cams, 45 carbs, early-pattern GC verniers, 42/38 valves. Estimated output 170bhp @ 7800rpm. Sadly 4-1 would not fit in car – a Sylva Leader – and was replaced by manifold beautifully made by the late Len Hartley.



15/16: Note clips retaining primaries to collector. Fiat external thermostat replaced with 74°C in-head type. Layout allowed good airflow around exhaust system – a crucial factor in maintaining low under-bonnet temperature. Cam belt cover sensibly retained – protects fingers, and cam belt from stones!



15/17: 4-1 primaries on this Mk 1 Escort/Fiat conversion, carefully shaped to equalize primary lengths and still fit available space. Early 124 1608 is fitted with sidedraught 40s (with top-mounted 124 distributor). Engine is mounted behind Escort crossmember, giving good front/rear weight distribution. Note lock tabs used only on early camwheels.



15/18: Torque curves of GC engine No 210, St II 2.1 (86mm bore) Fiat, IIIA cams, 43.5/36 valves, 11:1 CR, 45 DCOE (38 chokes); without silencer (1) and with (2). Overly small box (18" long, 8" x 4" oval, 2" ID) led to mild enhancement of torque (shaded) at 5,000 and 6,000rpm, with losses in the rest of the range. (It is unlikely that the small bore of the silencer was creating any useful pressure effect.)

15/19: Complete system on Claes Eklund's Bentec 7 comprises dual silencers often fitted on production cars. Beautifully finished car sports rare ex-124 Abarth Spider Cromodora wheels.

level, but the system can also provide the secondary benefit of damping out unwanted pressure waves in the tailpipe.

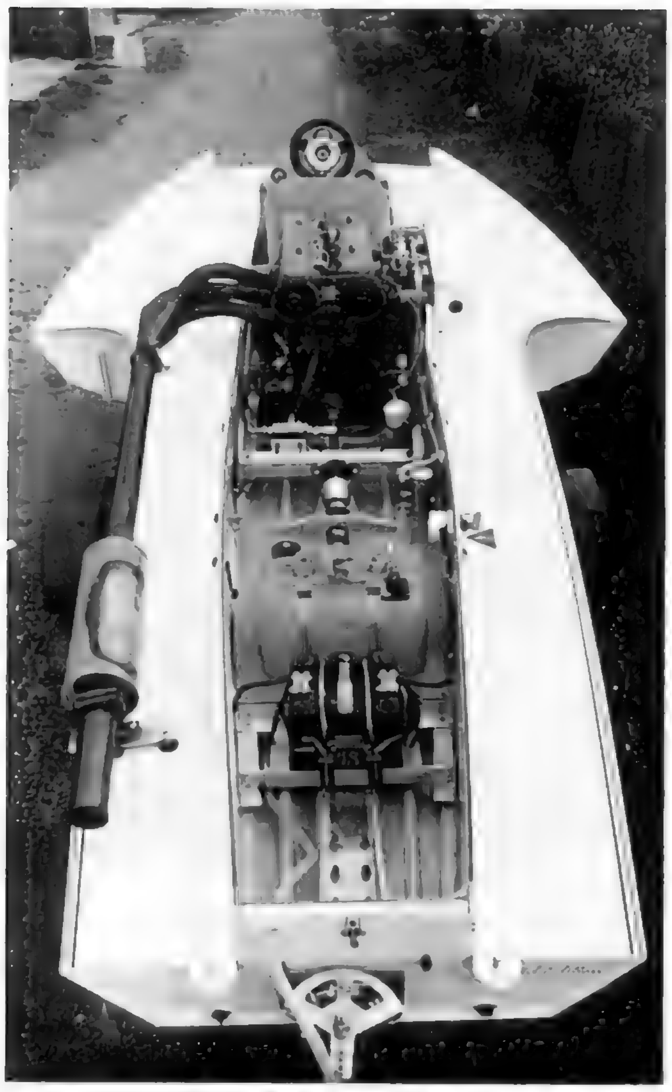
However, if the silencer is too small it will create the phenomenon known as back-pressure, which in all cases is highly undesirable. Back-pressure is frequently confused with wave action, which is an entirely different effect. Back-pressure results from the pressure of 'stalled' gas trapped in the pipe; obviously this inhibits the passage of exhaust from the engine and causes power loss.

Furthermore, if for example an engine is fitted for optimum mixture on a dyno, where the silencing system is normally extremely large and of negligible back-pressure, and is then married up to an unsuitable silencer, the resultant back-pressure will reduce the airflow into the engine and upset the jetting. For this reason, it is useful to carry out a rolling-road dyno test on the car after fitting if there is any doubt about the capacity of the system.

Back-pressure can be measured simply with a 0–10lbf/in^2 gauge. Readings can be taken at the manifold and either side of silencers. (5/21)

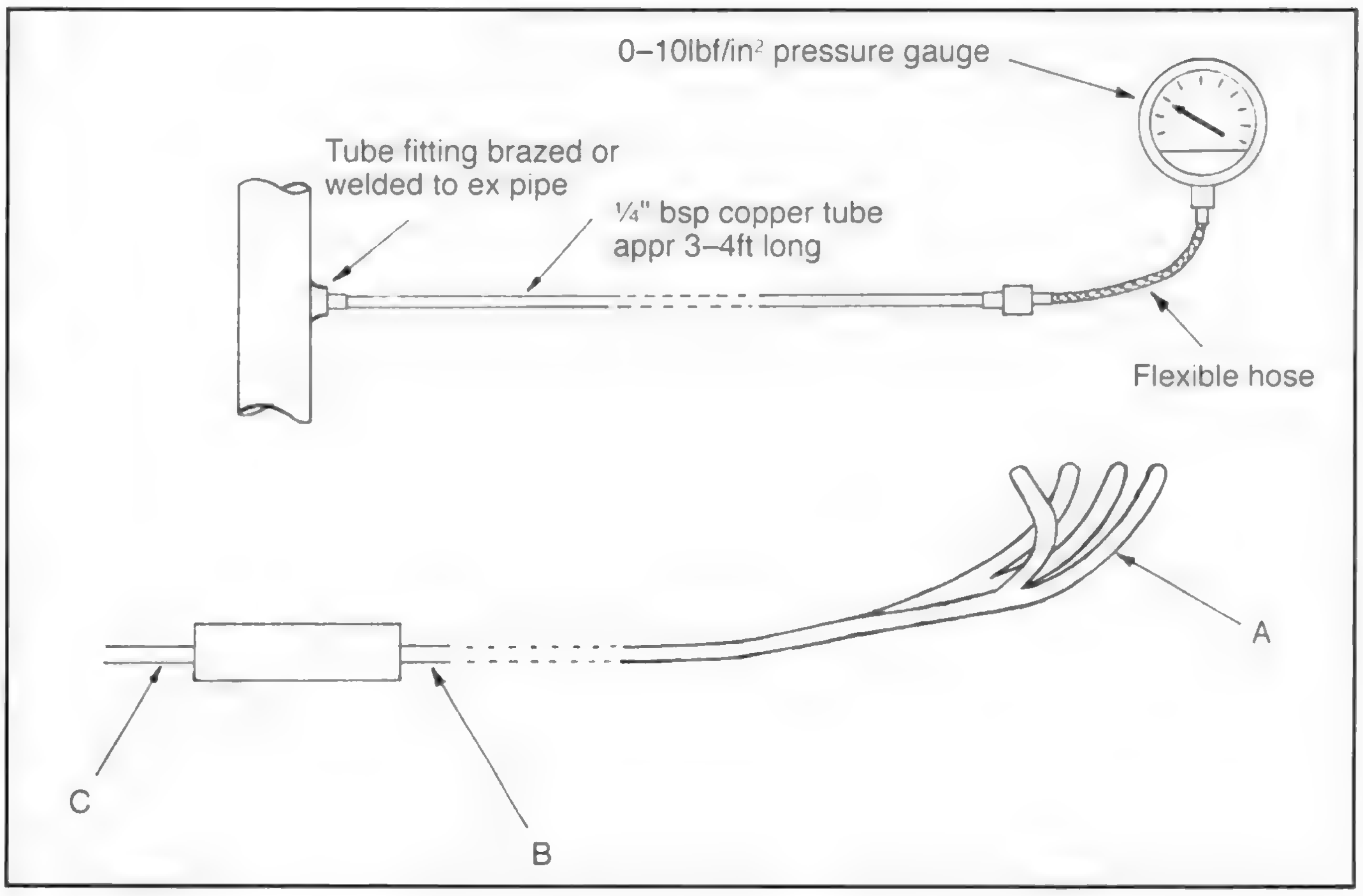
As a rough guide, a reading of 0–2lbf/in^2 means there is negligible back-pressure; 2–4 suggests improvements will lead to an increase in power; and 4+ warns of a serious back-pressure problem. Excessively tight bends, constricted pipes, sharp corners, and a silencer too small or blocked are some of the possible causes of excessive back-pressure. On a competition car it is common to use a single 'straight-through' silencer. It is feasible to utilize a silencer from another model, *eg* an Escort RS2000, provided it meets the noise criteria for the particular type of motorsport.

Turbocharged engines are inherently quieter than an equivalent normally aspirated engine since the sound energy of the primary wave is damped out by the turbine unit. The same is certainly not true of supercharged (*eg* Volumex) engines, which tend to produce a raucous 'tearing calico' rasp! For this reason, the

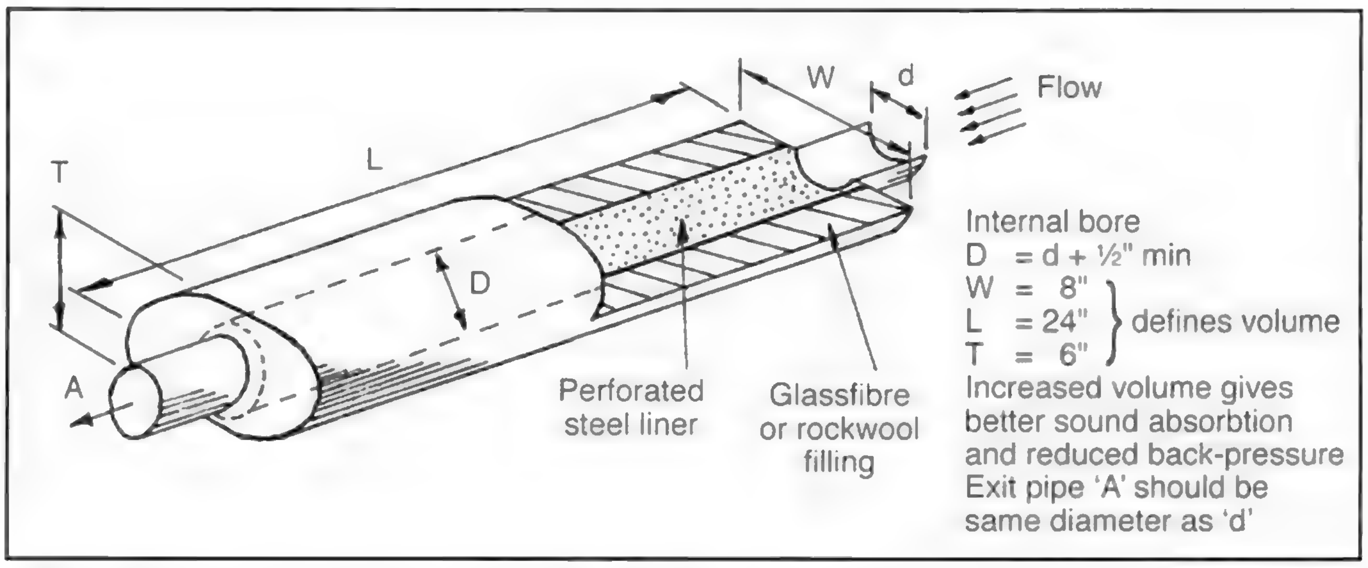


15/20: Formula 2000 hydroplane – Grand Prix racing on water – features St III race GC 1800 Fiat, 320° cams, 48 DHLAs (40 chokes), 11:1 CR, 44/38 valves, 4-1 manifold. Primary ID 37mm, tailpipe 56mm, single 70mm ID straight-through silencer.

Only RH-rotation propellers were available, so propshaft driven from crank nose. Dry-sump lubrication with oil tank located behind driver. With power output of around 180bhp this craft – half-boat, half-plane – broke Oulton Broad lap record formerly set by an Allison aero-engined hydro in Fifties. Flexible pipe on transom carries water under pressure from rudder to cool engine. Note prominent oil pressure warning light behind steering wheel.



15/21: Back-pressure measurement. Tap into exhaust system in primary pipe and either side of silencer(s) as shown.

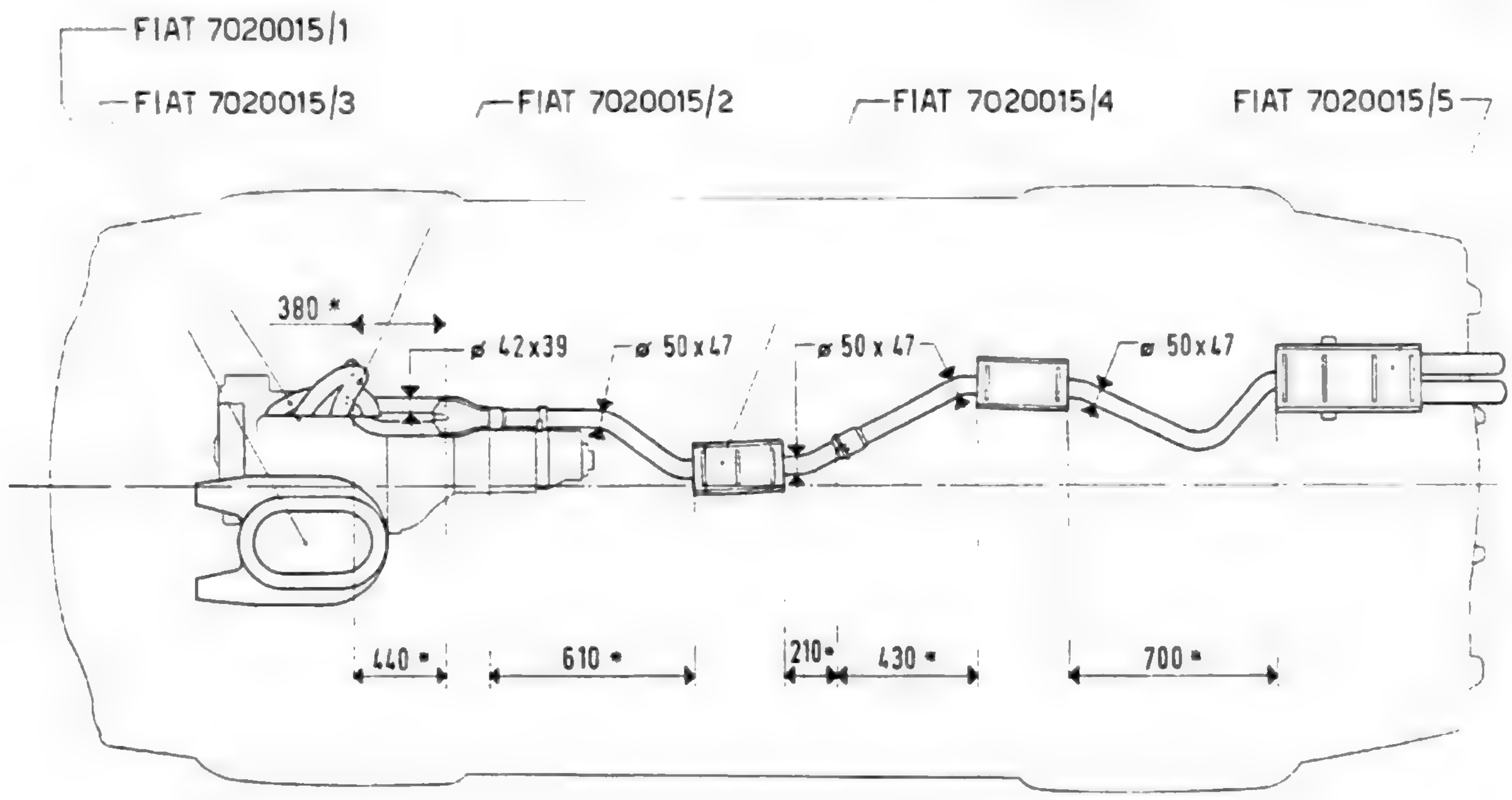


15/22: Section through typical competition silencer, St II/III (oval).

turbocharged engine can use a smaller silencer than an n/a engine of equivalent power. A resonator would be useless, since the reflected wave would be lost in the turbocharger. The same principles of manifold design apply to both engines, though 4-1 tends to be preferred for turbos for space reasons. Clearly, the consideration of primary wave effect is not nearly so crucial to the effective filling of the cylinder since the fresh charge is forced in at pressure above atmospheric once the throttle opens. A considerable amount of back-pressure exists in the exhaust manifold on turbos at part-throttle because of the restriction of the turbo.

During WW2 it was discovered that sleeve-valved aero engines were particu-

15/23: Leif Bengtsson's Bentec 7 fitted with early single silencer, 2½"ID, made from carbon-fibre. Original Swedish Sports Car race noise level of 110 dB @ 3500rpm was lowered to 100 dB in 1993 and silencer had to be enlarged to increase baffle area. Built by Leif himself, these cars are constructed to highest standards, as one would expect from an engineer who makes machine tools for Volvo!



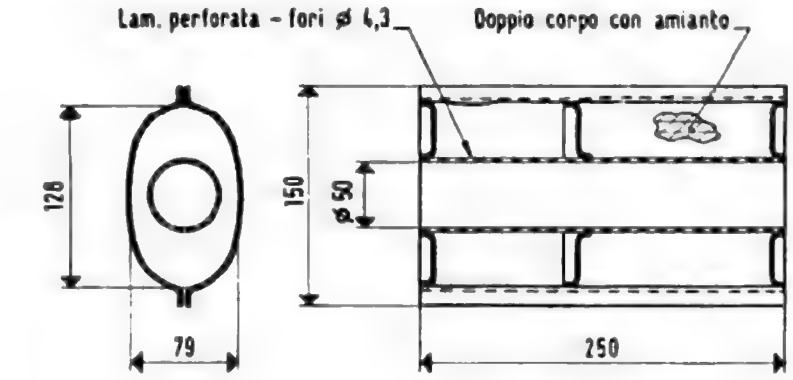
* Lunghezza sviluppata

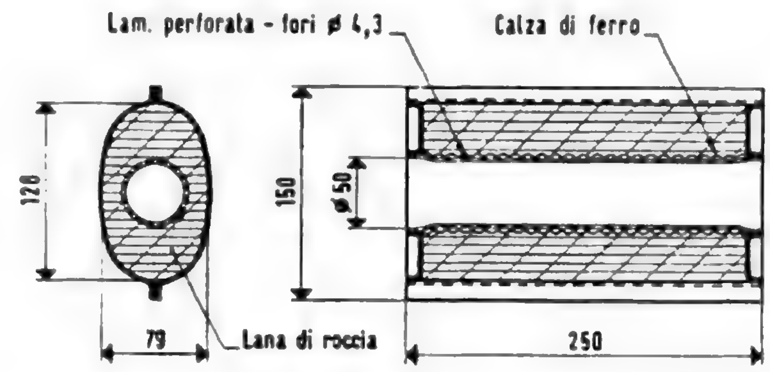
Valori del livello sonoro:

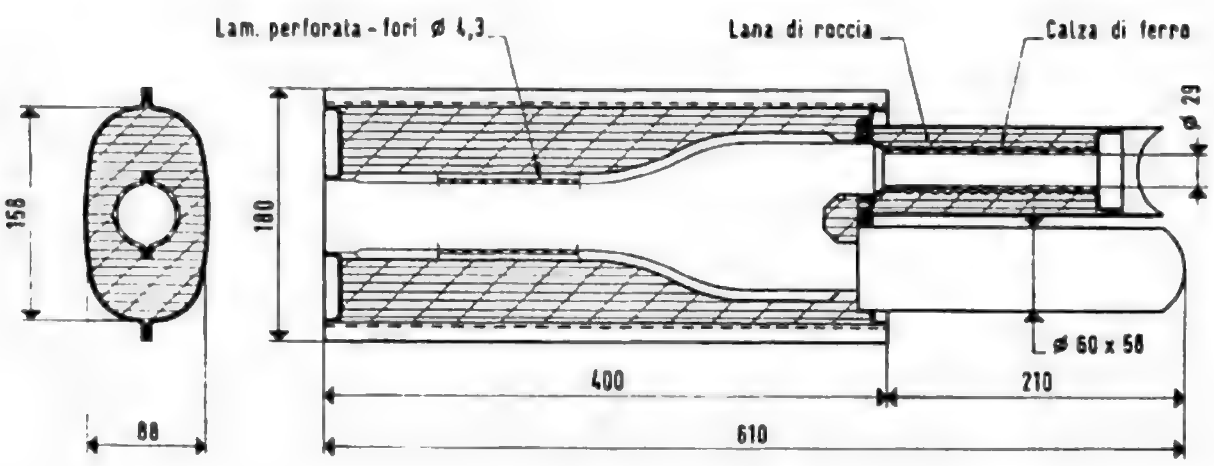
— veicolo in moto: **78 dB** (A); velocità stabilizzata prima dell'accelerazione: **50** km/h;

— veicolo fermo: **76 dB** (A) a **4650** giri/1' del motore;

— livello sonoro per veicoli usati: **85 dB** (A) « E »; regime di controllo: **4650** giri/1'.







15/24: Extract of homologation papers for Fiat Abarth 124 Sport Rally (Spider) shows layout of exhaust system for 128bhp (DIN) 1756 engine. Consists of 12" primaries, 24" secondaries and triple silencer system. Silencers consist of basic casing with perforated steel liner packed with rockwool. Note very low (85 dB (A)) noise level at 4650rpm. Centre box has diameter reduction of approx 4mm to force noise absorption material to function.

15/25: Gorgeous Lancia Transformer Stratos replica belonging to Phil Jordan; VAG 'supersalesman' and himself a qualified motor engineer, he built car over period of two years. Original exhaust system featured paired primaries and split secondaries running into single large-bore silencer with two exit pipes. Chronic exhaust interference between paired cylinders 1 and 4, 2 and 3 was caused by use of very short (10") primaries. Reverse flow from primary was remaining in pipe at low rpm, re-entered cylinders and snuffed out combustion at irregular intervals. Later system extended primary pipes to 16" and cured problem. Exclusive car by outstanding Frant, Sussex-based designer Gerry Hawkridge rapidly achieved wide acclaim for chassis/body strength, simplicity and close adherence to original design. This particular model features GC engine No 53; 2l Lancia, ported head, standard cams, 9.6:1 CR, 45 DGLS (36 chokes), developing 155bhp @ 6300rpm with 142lbf ft torque @ 3800rpm. Very broad spread of torque despite modest top-end power results in phenomenal acceleration through gears and top speed potential of well over 125mph.

15/26: Business end of Peter Luxford's dramatic Martini-liveried Volumex-engined Stratos replica. Pedestrian number-plate belies ferocious performance. GC engine No 153 develops 170bhp at wheels on 16" tyres, at least 215bhp at flywheel. Unlike turbos, supercharged engine develops serious usable torque right from tickover. Exhaust system is 4-1 with Porsche Turbo silencer – real treat for ears on full song!

larly suitable for high back-pressure operation, but poppet-valved engines suffer very adverse reverse flow at low turbine speeds, *ie* when the exhaust gas volume and velocity are minimal. This is usually inhibited on production turbo engines by keeping valve overlap and exhaust valve lift to a minimum.

RAC NOISE LEVEL REGULATIONS

Noise levels for various forms of UK motorsport are laid down by the RAC Motor Sports Association; their regulations from the *1995 RACMSA Year Book* (the *'Blue Book'*) are reproduced here. Competitors are advised to study this carefully since failure to conform to the regulations (see especially 13.17.7) can easily result in exclusion from an event, or a track day!

13.16 Exhausts

13.16.1 Have the EXHAUST SYSTEM isolated from the driver/passenger compartment (*eg* beneath the floor or secured in casings of solid material).
13.16.2 Have no part of the EXHAUST SYSTEM protruding to the rear of the bodywork more than 15cm.
13.16.3 If a Racing Car with rear aerofoils, do not have any EXHAUST PIPES extending rearwards beyond the aerofoil.
13.16.4 If a Racing Car without aerofoils, do not have EXHAUST PIPES extending more than 60cm beyond the rear wheel axis.
13.16.5 If a Rear Engined Single Seater Racing Car, have the EXHAUST OUTLET between 4cm and 60cm from the ground.
13.16.6 Have all EXHAUST OUTLETS terminating behind the mid-point of the wheelbase of the vehicle and outside the bodywork periphery in plan view. Side exhausts not to protrude more than 4cm. Cars built before 1961 are exempt from these requirements.

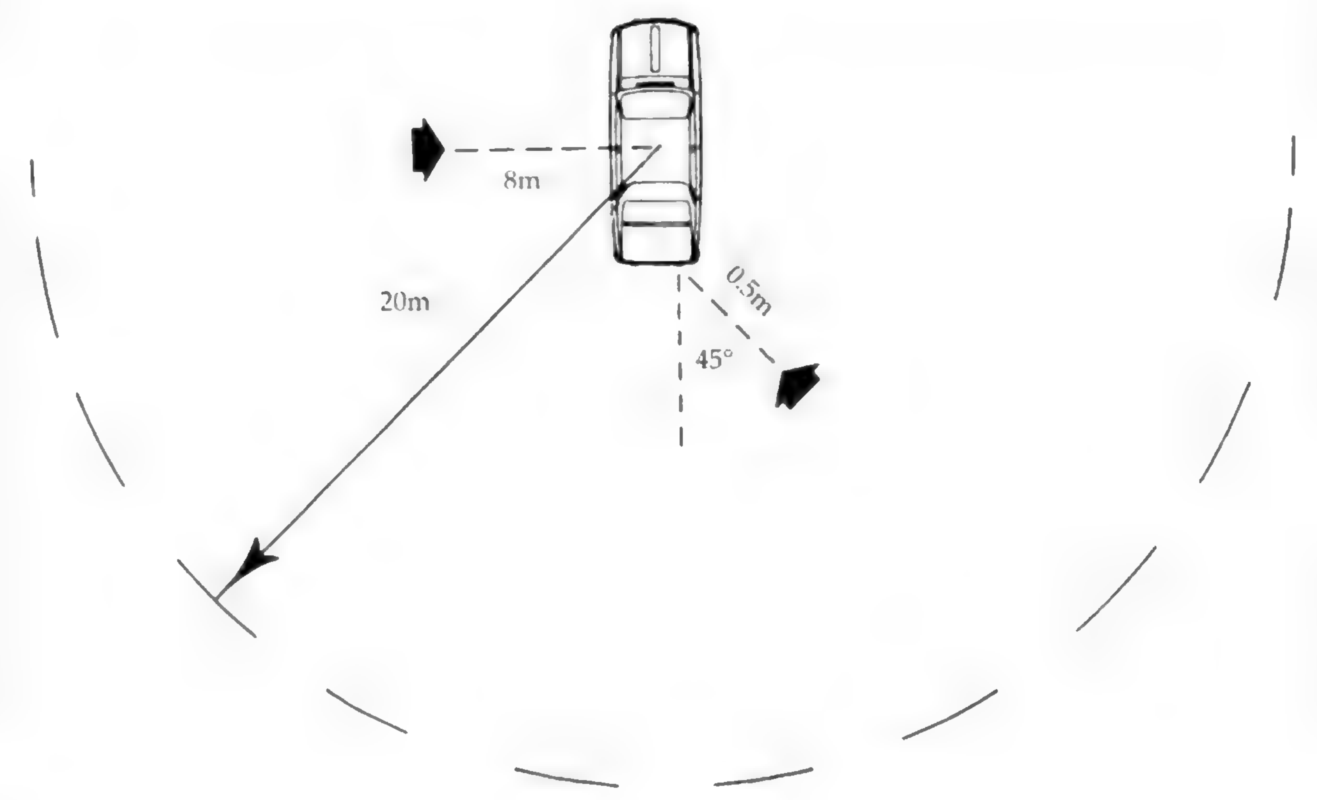
13.17 Silencing

The reason for Silencing (SOUND CONTROL) is to reduce environmental impact and to keep Motor Sport running. Recent Environmental Protection legislation has increased the pressure on activities generating noise and Local Authorities have the power to suppress any noise source deemed to be causing a nuisance. Our system of control is acceptable to most Environmental Bodies and must be considered as part of ELIGIBILITY to compete in events.

13.17.1 All competing vehicles are subject to MANDATORY SILENCING, unless a specific waiver for that Class, or Formula is granted. Where specified as mandatory, a silencer must be used, irrespective of the exhaust sound generated without it.
13.17.2 The following waivers are granted:
FORMULA 1, FORMULA 3000, FORMULA 3, when competing in current FIA Championship events, and Drag race vehicles.
13.17.3 Silencing is not mandatory for the following categories of vehicle but is strongly recommended and may be made mandatory in the SR [Supplementary Regulations] at the request of the Circuit/Venue owners:
VETERAN CARS built prior to 1905
EDWARDIAN CARS built prior to 1919
VINTAGE CARS built prior to 1931
POST VINTAGE THOROUGHBRED CARS built prior to 1941

HISTORIC RACING CARS – FORMULA 1 built 1939 to 1965
HISTORIC SINGLE SEATERS built to comply with International Grand Prix, Voiturette, Formula 2, Formula 3 or Formula Junior regulations 1931 to 1965.
13.17.4 Special regulations will apply for British F2 and F3.
13.17.5 Temporary Silencers, By-Pass pipes or the inclusion of temporary parts to achieve silencing requirements are prohibited. Officials may refuse to carry out Sound Checks on vehicles utilizing temporary parts in exhaust systems. Organizers are empowered to exclude in such situations.
13.17.6 Due to changes in EU Regulations, sound levels may be reduced from 1995.
13.17.7 Circuit/venue owners/organizers may impose additional restrictions in SRs.

The following table gives alternative distance readings. (Noise measured in dB(A).)



15/27: Sound test requirements – angles and distances.

	0.5m	2.0m	8.0m	16.0m	
Section 'A'	110	98	86	80	CAR RACE AND RALLYCROSS MAXIMUM AT 3/4 MAXIMUM RPM
Section 'B'	113	101	89	83	HILL CLIMB & SPRINT MAXIMUM AT 2/3 MAXIMUM RPM
Section 'C'	108	96	84	78	AUTOCROSS, AUTOTEST, STAGE RALLY, TRIALS, CCV MAXIMUM AT 5000 RPM
Section 'D'	102	90	78	72	ROAD RALLY MAXIMUM AT 5000 RPM

13.17.8 Sound Test Requirements. Measurements will be made at 0.5m from the end of the exhaust pipe with the microphone at exhaust outlet level at an angle of 45° with the exhaust outlet. Where more than one exhaust outlet is present, the test will be repeated for each exhaust and the highest reading will be used. In circumstances where the exhaust outlet is not immediately accessible, the test may be conducted at 2.0m from the centre line of the vehicle at 90° to the vehicle, with the microphone 1.2m above the ground. Measurements should be made outdoors with no LARGE reflecting objects (*eg* walls etc) within 3.0m (in the 0.5m test) or within 10.0m (in the 2.0m test). Background sound levels should be at least 10dB(A) below the measured level.

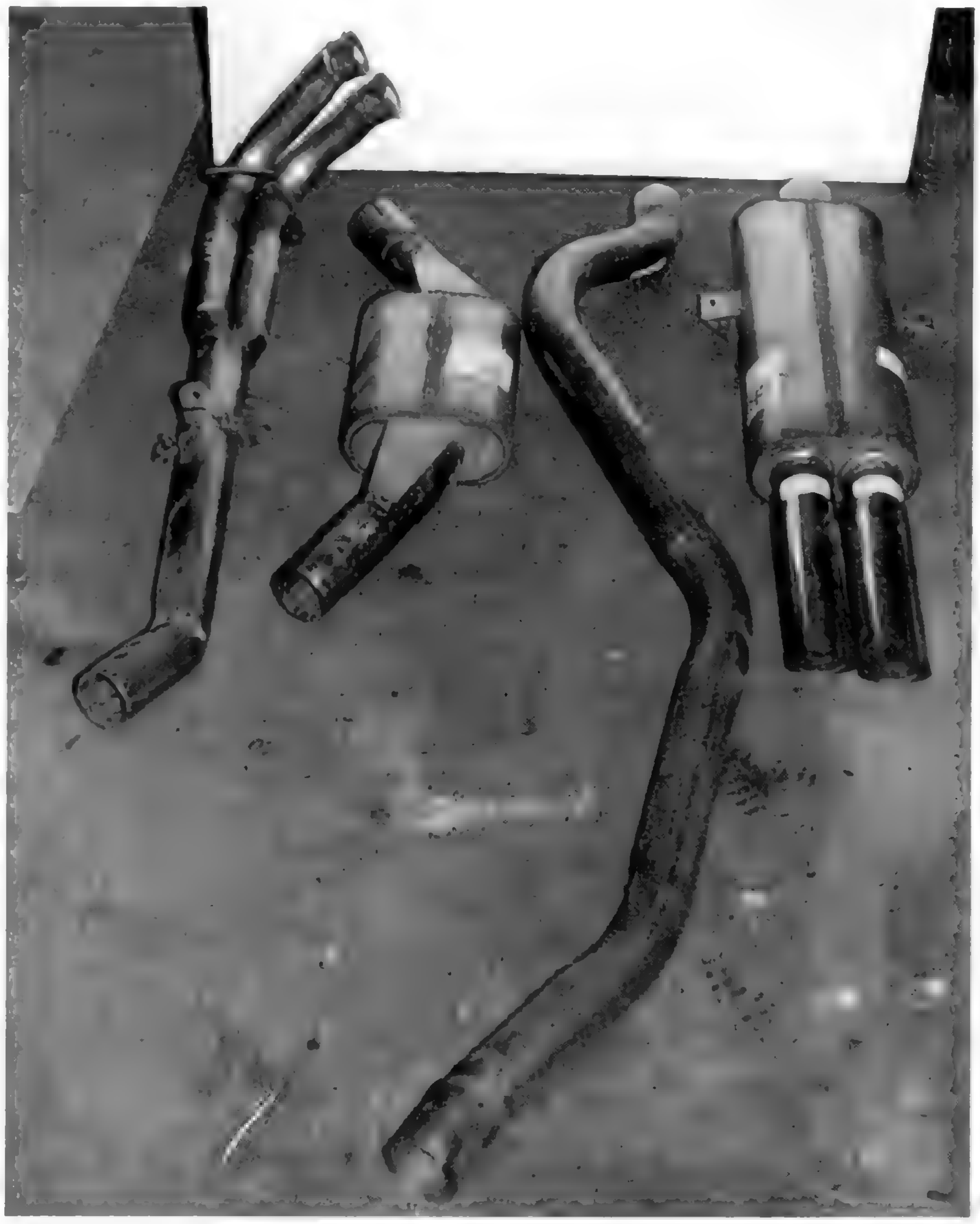
With distances from 2.0m to 8.0m it is necessary that there be a minimum of 20.0m radius open flat space around the vehicle.

Where possible measurements should be taken as close as possible to the vehicle, at the defined distances, to avoid background noise.

Generally it is impracticable to take measurements over 8.0m as the background noises create problems with accurate and steady readings.

Pre-1939 cars in Sections 'C' and 'D' should run engines at two-thirds maximum RPM.

The 8m and 16m locations of the microphone, for practical purposes, can be considered to be 7m and 15m from saloon car bodywork. The measurement can be made from either side of the car. The highest reading registered being the one needing to



15/28: Stainless steel exhaust system for author's 124 CSA by P D Gough. One silencer was left out with only marginal increase of noise level, but a much sportier sound! It was copied from old system (as opposed to being made on car) and fitted like a glove!

comply with the MAXIMUM NOISE requirements.
13.17.9 Sound testing should be carried out BEFORE taking part in any competition. The time and location of sound testing should be advised to competitors prior to the event.
13.17.10 It is stressed that all participants in motor sport, competitors, officials, marshals, etc, should be aware of, and protect themselves from, noise.
(Data kindly supplied courtesy of the RAC Motorsports Association.)

15/29: Engine bay of Julian Sudano's US-spec Spider illustrates that, despite all theory, 'if it works, it works!' 2l, 9.8:1 CR, fitted with 40 IDFs, rally cams, uses standard manifold connected to single secondary pipe; against odds, seems to work fine! Now-hard-to-get top-mounted distributor was found on 124 BC 1608, US Spiders and Argenta injection version.

CASE HISTORY No 7

Owner	Paul Thomas
Type	2/131, 84.6mm bore (2024cc) St IV GC Eng No 211
Use	National Hot Rod
Tested	Warrior Automotive, June 1994
Rig	Superflow

Specification:
Forged pistons 11:1 CR, 130 TC head gasket, race bolts.
45 DCOEs, 40 choke, GC sidedraught manifold.
GC IIID cams.
46/40 race valves, triple springs, alloy caps.
Race wet sump plus Accusump.
Ultra-light steel flywheel, balanced crank/flywheel assembly.
Direct-drive water pump (using V-belt – led to problems in 1995).
4-1 manifold 22in.
BP9 EGV NGK plugs.
Supergreen (high-octane unleaded) fuel.

CH7/1: GC engine No 211. Note latest pattern 'full-form' big-wing sump, oil take-off plate. Filter was ITG with single backplate. Coolant outlet elbow was modfied from standard 131 type. Distributor is Bosch electronic from Lancia Beta 2l. Accusump was deemed a wise precaution due to tight, high-g turns in Hot Rod. Model is 1qt version which was fitted with a solenoid valve on actual installation.

This engine was a rebuild of engine No 189, which had been badly damaged during racing when a stone caught under the cam belt. The opportunity was taken to replace the pistons with forged items as feedback from Irish Hot Rod driver Tom Casey, whose car was fitted with a telltale tacho, indicated that Paul Thomas' engine was almost certainly revving to 7800–8200rpm!

Initially the engine was installed directly in the car and attempts were made to re-jet the carburettors by track-testing. This was not successful and no amount of 'tweaking' and testing would make the engine pick up below 4000rpm (even when unloaded).

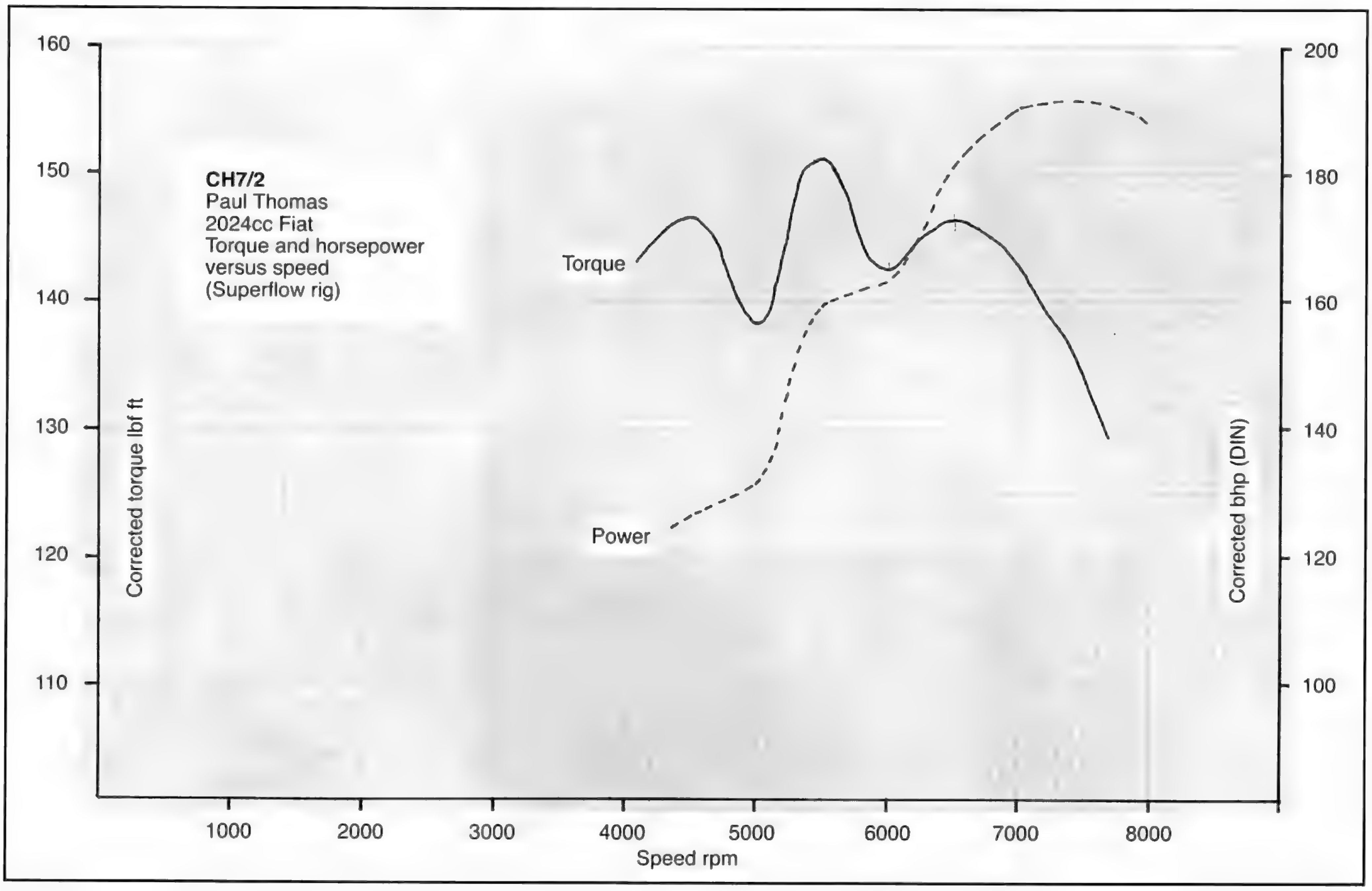
Following a disastrous race showing, the engine was removed and taken back to GCT for further examination. A simple back-to-back test with a longer 4-1 manifold cured the problem instantly. The manifold supplied (direct from the manufacturer, unfortunately, to Paul Thomas) had only 12in primary lengths and was clearly causing chronic interference between cylinders.

The two hours of testing and racing had led to ring/bore damage – because at the track it was felt that the problem had been caused by over-lean mixture and the jets had been changed for progressively larger ones, whereas in fact the problem was actually back-pressure and the excessively large jets led to bore washing over 4000rpm.

No such problems were encountered with the rebuilt engine using the 22in 4-1 manifold. The engine was run-in on Warrior's Superflow dyno for two hours and performed flawlessly throughout the test session. The final results are recorded on the graph. On the flexibility test, the engine pulled full-throttle from 3500rpm.

Final jetting was	170 main
	F16 emulsion tube
	175 air corrector
	60F9 idle jet
Ignition advance	36° @ 5500rpm

[*Author's note:* This engine was eventually 'lunched' in May 1995 when the V-belt water pump drive flew off and caught in the cam belt! <u>QED</u>]



CHAPTER 16

TESTING

GENERAL

An engine which has been fitted with new piston rings must be subjected to a running-in period to allow the rings to bed in. Failure to carry out this phase will lead rapidly to bore and ring damage and chronic power loss. During running-in it is *vital* that mixture, ignition settings, oil pressure and temperature and coolant temperature, etc, are correct.

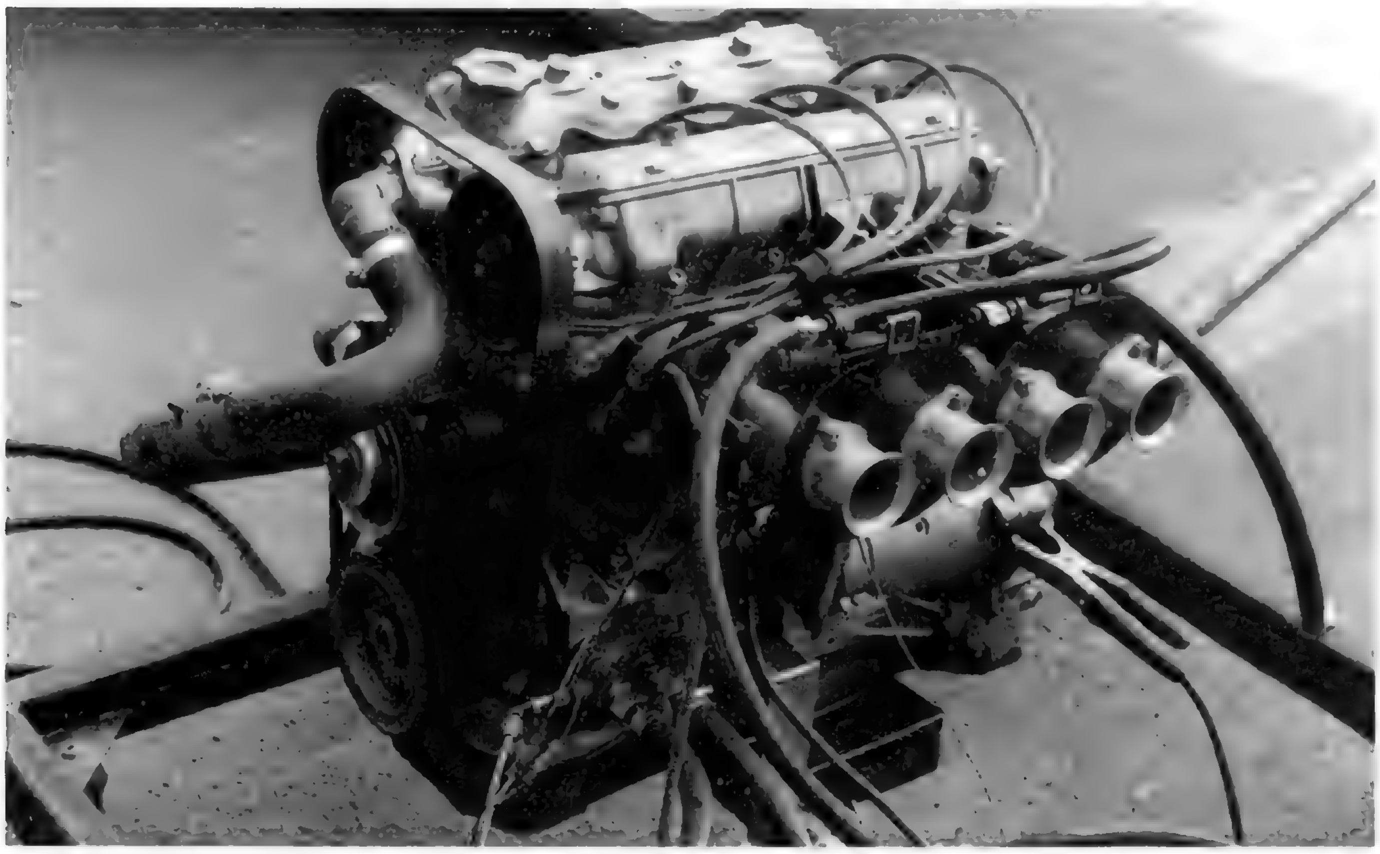
For example, if the mixture is chronically over-rich, bore washing will occur (usually when the CO content exceeds 6%), or if it is over-lean (less than 4.5% under load) pre-ignition or overheating may result. Overheating will similarly result from incorrectly set ignition timing. Clearly, an engine which has been rebuilt to standard specification should not require alteration to the fuelling, but a modified unit must be recalibrated prior to, and after running-in. Whether the engine is set up in the car by road-testing, a rolling-road, or on a bench dynamometer (dyno rig), the procedures should be the same, *ie:*

1 Preliminary checks.
2 Running-in.
3 Load-testing.

Preliminary checks

Ensure the valve clearances are correct, pour cam start-up lubricant over the cams (new cams/shims), ensure the sump plug is tight, fill the cam boxes with oil and fill the engine with oil. Remove the plugs. Fill the cooling system with water, ensure the drain plug is tight. (Do not add antifreeze – except in winter! – at this stage in case there are any leaks and the system has to be drained again.)

Set the distributor at the static advance setting (remember that when the cams are on their TDC marks, No 4 is firing). If race carbs are fitted, set the carb balance approximately, idle mixture screws 3–4 turns out. Set the idle speed screw in to make light contact with throttle quadrant then add one turn. Activate the fuel pump and check for leaks, then disconnect the main power feed to the coil or ignition pack, open the throttles and crank up the oil pressure. If the pressure does not come up within two or so 10-second cranking sessions, remove the oil filter housing and pump the engine oil into the main feed gallery from the pump. A dry-sump pump should be spun with a speed wrench or power drill.



16/1: Ray Carden's 2l rebuilt grasstrack engine undergoing hot-run tests prior to installation in car. Main jetting had already been proved on earlier engine by rolling-road tests. Carbs are 45 DHLA, 38 choke. Simple rig allows compression tests, balance and idle mixture, ignition timing checks to be carried out. Cam boxes are Fiat – only covers are Lancia. Note large-capacity big-wing sump with rear well for Starlet set-up. No alternator required, so water pump is driven direct from crank via V-belt (later changed to tooth-belt drive). No thermostat is fitted at this stage.

In the car it is advisable to pre-fill the cooler, remote filter and oil lines to avoid cranking the engine for long periods to raise oil pressure – since damage to the bearings could occur. Reconnect the ignition, prime the engine by opening the throttles fully three to four times (this will actuate the pump jets). Start the engine. (It may be necessary to swing the distributor either way at this stage to get it to run.) When the motor fires, hold the throttle slightly open and hold the speed at around 2000rpm for about 4 minutes to allow the cams to bed in (if new) and the engine to warm up.

Check the oil pressure; when the engine is warm, allow the throttles to close and set the idle speed screw to allow a tickover speed of 850–1000rpm.

With a strobe light, check the ignition advance, both at tickover and at maximum advance *(see Chapter 11 for settings)*. Balance and idle mixtures can now be checked *(see Chapter 10)*. Obviously, check for leaks of oil and coolant, plus the exhaust system can be checked for sealing. Any leaks will become worse – not better – so rectify at this stage. Do not be tempted to tighten the cam box or head bolts with the engine hot: this must be done with the engine stone-cold. When the engine has been hot-run for 10–20 minutes, it is worth carrying out a compression test. (It is not vital to run the engine under load to do this initial check.)

Isolate the ignition as before, remove the plugs, fit the pressure test gauge, hold the throttles wide open and crank the engine for approximately 10–12 revolutions. Although the rings have not yet

PRELIMINARY CRANKING COMPRESSIONS		
CR	COMPRESSION lbf/in²	
9.0:1	175–185	Std cams
9.6:1	185–195	St II–III cams
10.0:1	190–220	Depending on cam duration
11.0:1	185–200	St III – race cams

bedded in, reasonable compressions should be obtained since the engine and oil are hot, and the interference seats should now be at optimum seal.

Long-duration cams give a lower compression than short since the inlet cam is held open for a considerable period after BDC and the mixture tends to be pumped back out of the inlet tract. Variance between cylinders should not exceed ±10%. The pressure reading on the gauge should rise quite sharply to around 120lbf/in² on the first compression and then the rate of increase will diminish. This is because of leakage taking place past the rings at high pressure with no combustion gas force acting on them. The battery must be in good order, or the readings will be meaningless. If any of the readings are unacceptably low, proceed with fault finding at this stage, *eg* shims too tight, incorrect cam timing, throttles out of balance. A leak-down test may be advisable.

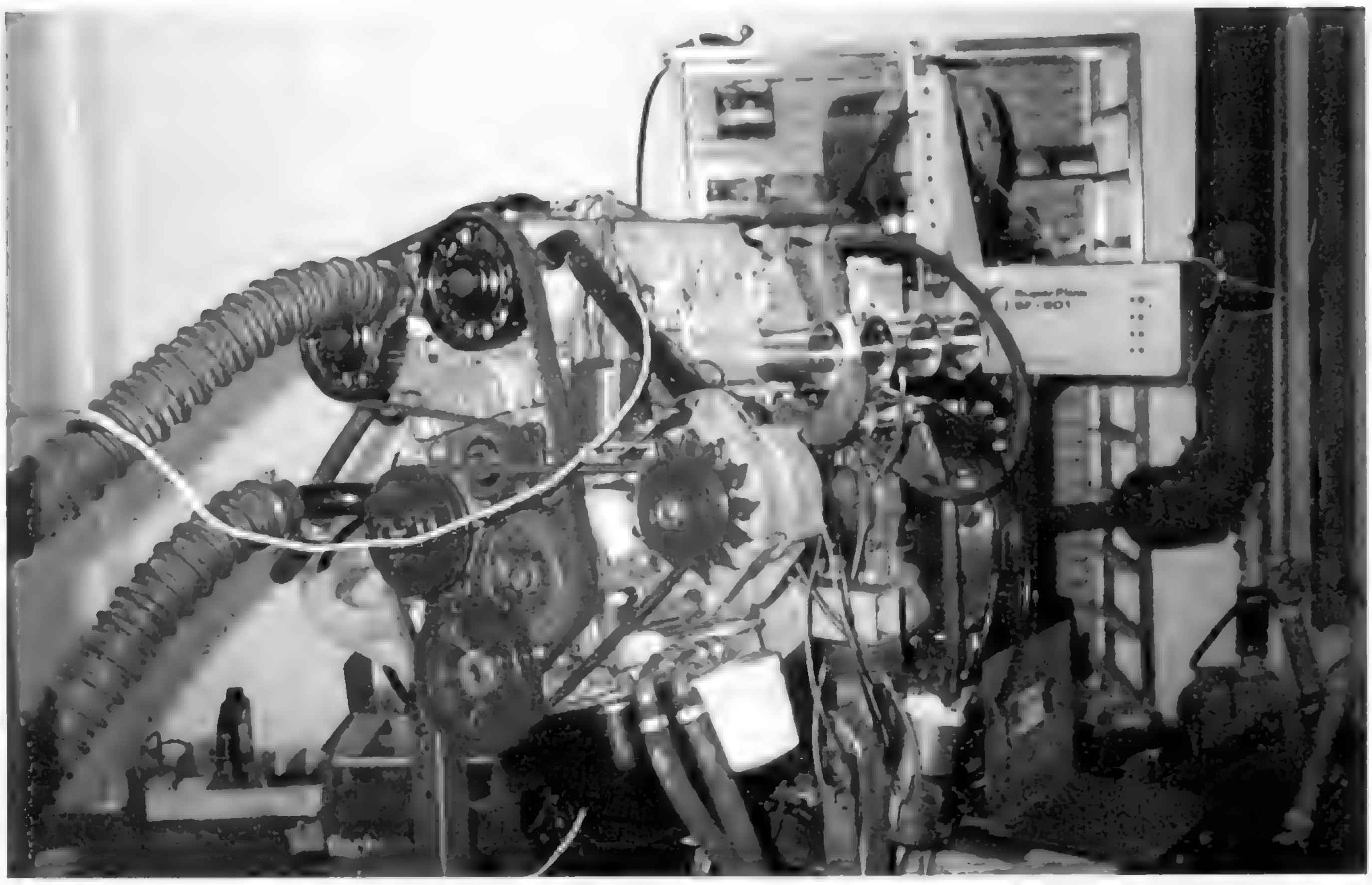
Assuming that all is well at this stage, the engine is ready for running-in. Note that it is not possible to run-in without loading-up the engine because adequate ring-to-bore friction will never be achieved at the light throttle openings which can be achieved 'off-load'. A load must be applied and the throttle opened progressively towards maximum during running-in to ensure full pressure on the rings.

BENCH DYNO TUNING

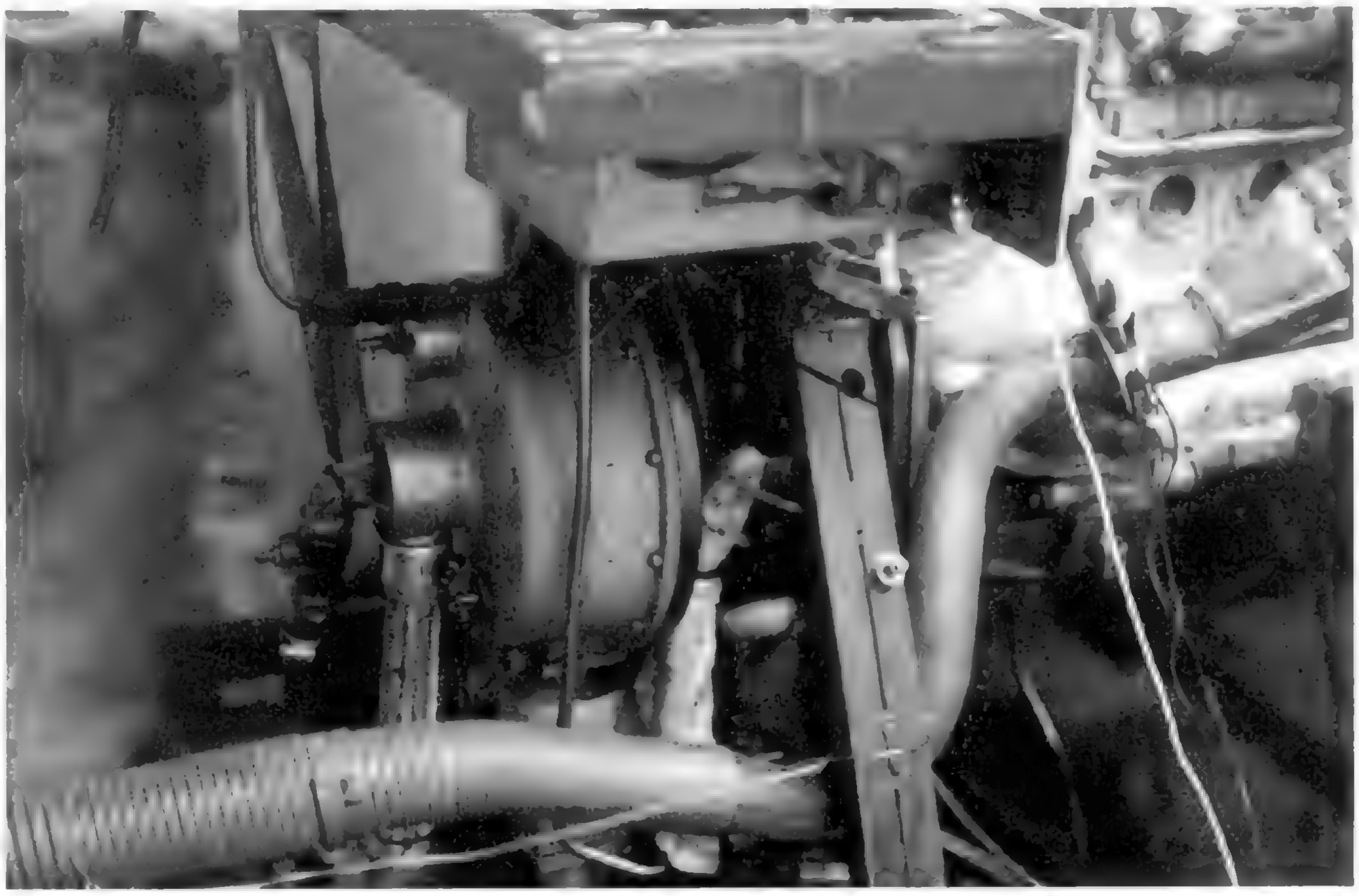
Running-in and load-testing

Setting up by this method represents the safest and most accurate way of running-in and testing an engine. A well equipped dyno cell will not only provide accurate measurement of output and mixture levels, but also safe monitoring of oil pressure/temperature, coolant temperature and test cell temperature, with appropriate shutdown procedures should any of the readings fluctuate dangerously. It goes without saying that any system of engine setting-up or testing, however sophisticated, is only as good as the operator.

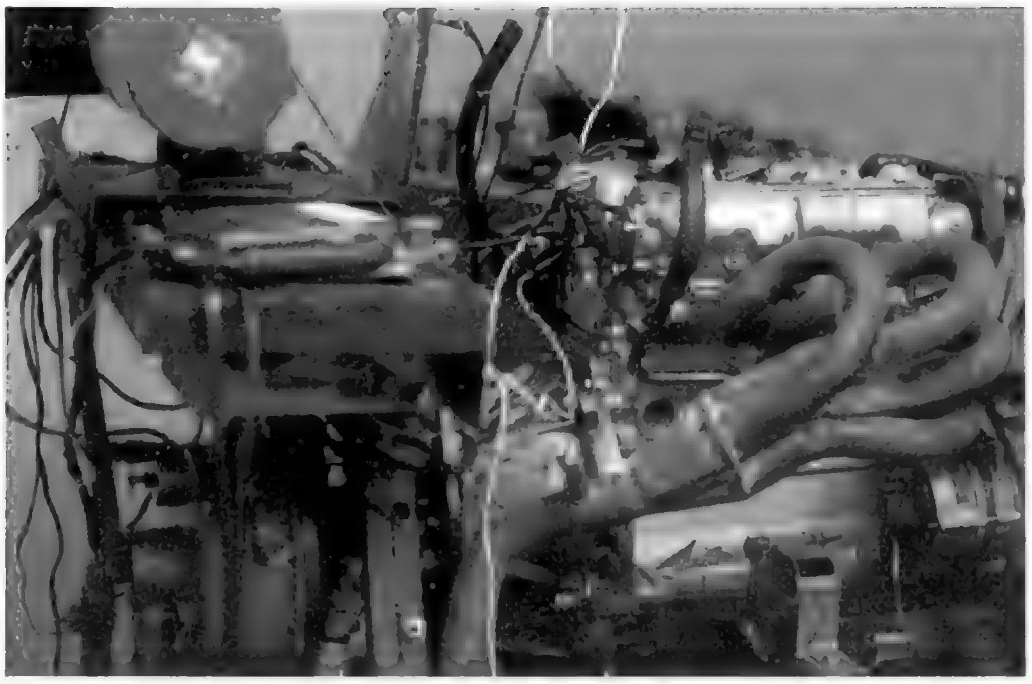
Unless the engine is being tested with 'mapped' ignition and/or fuel injection



16/2: Nigel Peyton's 2l Lancia on Superflow rig at Warrior Automotive (see Case History No 9). Large hoses to left of shot carry coolant. Hose between carbs is breather, which would be connected to catchtank in car. Sandwich block on oil filter housing feeds oil to heat exchanger controlled by rig computer to hold oil temp at 85°C. Exhaust gas is carried in large duct (far right corner) to massive silencer above dyno cell. Although not visible here, trapdoor above dyno rig houses one of two large-capacity cooling fans which control cell temperature. Pumps at right feed water to and from hydraulic dynamometer via cooling tank. (GC Eng No 88.) Thermostats must not be used on dyno tests.



16/3: Close-up view of dynamometer. This Superflow rig at Warrior Automotive Research can handle up to 1000bhp! Unit essentially operates by the 'shear' effect of the water inside the casing created by the rotation of the dyno rotor, with gate valves to control flow of water. As load is applied, twisting of dyno is recorded by electronic strain gauge, converted to Nm torque by cell computer.



16/4: Colin Haggett's 2l rally Fiat engine under test. Lambda-Sond fitted downstream of 4-1 collector. ECU for Weber-Marelli injection system sits on bench at left. Note individual coil units on plugs – engine has no distributor or coil.

16/6: Renowned Ford/Cosworth engine builder and dyno test expert Tim Swadkin at controls of Superflow rig at Warrior Automotive. Tim's hand is on throttle control, load is applied automatically by computer. When rpm reaches pre-set limit (eg 4000rpm) for start of power-run, Tim will initiate computer 'step test' at 500rpm intervals; computer will apply load automatically and record results. Analogue dials give quick visual checks of speed, load, fuel flow. Laptop computer at right is to programme Weber-Marelli ECU chip for Colin Haggett's engine. Small unit above console is Lambda-Sond dial, indicating mixture strength lean-rich.

and settings made relative to throttle position, dynamometer load tests are always carried out at full-throttle. The load is fed in progressively and the throttle opened simultaneously. The engine speed is then varied by altering the load; maximum load at full-throttle will correspond with maximum torque. At higher rpm, the torque will diminish, but owing to the relationship:

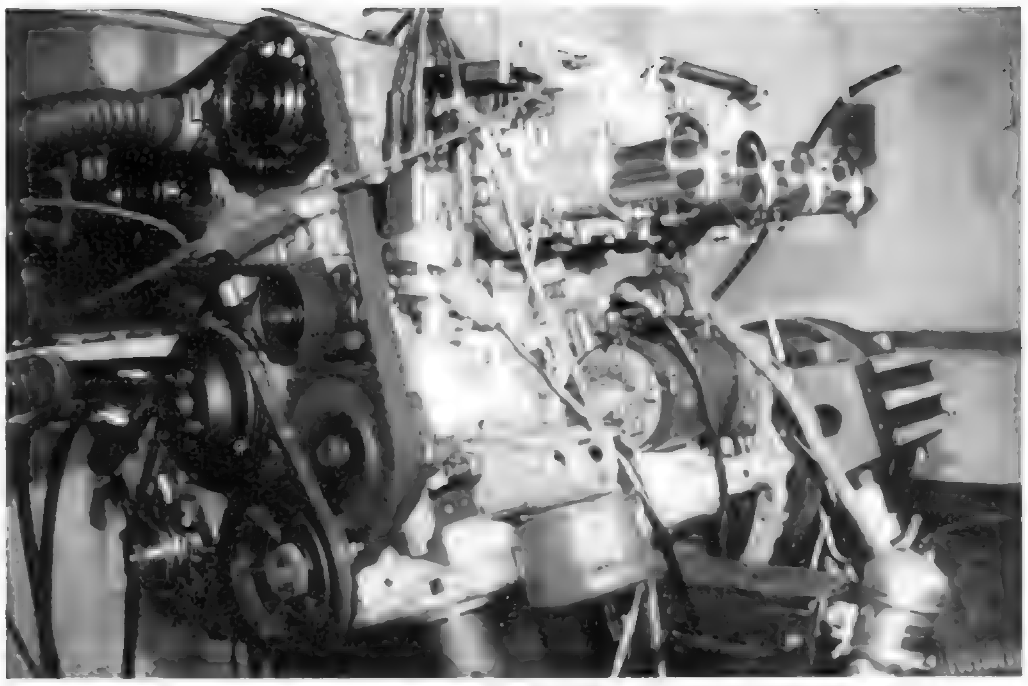
$$\text{Power (bhp)} = \frac{\text{Torque (lbf ft)} \times \text{Speed (rpm)}}{5250}$$

or (metric):

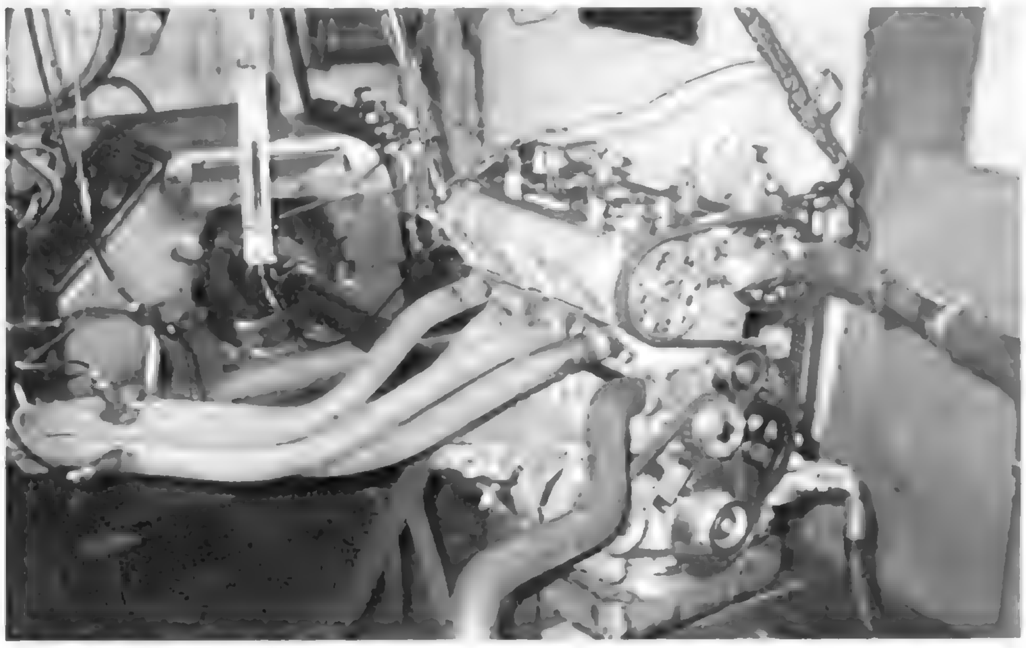
$$\text{Power (kW)} = \text{Torque (Nm)} \times \text{Speed (rpm)} \times 2\pi$$

the power will continue to climb until a point is reached where even though the rpm is high, so little torque is produced that marginal *usable* power is produced.

This running-in cycle should consist of approximately 2–2½ hours of varied speeds and loads. A warm-up period should be observed to bring coolant and oil up to temperature.



16/5: Dave Light's 2l Lancia on Go Power hydraulic dyno rig. Although not as accurate as Superflow rig, this dyno is still useful for back-to-back tests. Unlike Superflow rig, which measures mixture via bsfc (fuel flow meter), which stabilizes and gives accurate readings within seconds, this rig required plug checks to confirm mixture. This method requires sustained full-throttle testing. Lack of oil heat exchanger meant that full-throttle power runs had to be of fairly limited duration. Slim oil filter is from Volumex. (GC engine No 110 – see Case History No 8.)



16/7: Jonathan Douglas' special development 1700cc Fiat on the dyno at JE Engineering (see Case History No 4). Machining of 'chopper' and toothed belt pulleys took hours at GCT! Exhaust is a model of what a 4-1 should look like. Carb is Ford V6 Essex. Apart from early trouble with leakage past wire rings under solid copper gasket and combustion pressure crack in No 1 chamber, engine performed perfectly during countless power runs on detonation limit (13.2:1 CR on high-octane unleaded fuel) to optimize ignition.

If dyno connection is via propshaft as seen, alignment and balance must be flawless or shaft will shear in minutes. Note extension pulley made from blank and fitted to crank nose with standard nut. Alternator and water pump pulleys were similarly adapted. On a high-revving engine, there is no point in running high-speed ancillaries; toothed-belt drive is more efficient due to no-slip anyway.

16/8: Lumenition Lambda-Sond sensor.

On a competition engine, warm-up plugs approximately two to three grades hotter should be used to prevent fouling during this phase.

After the engine is rigged and warmed up, checks of idle mixture, balance and timing should be undertaken. A typical running-in sequence might then be:
(2l Fiat St II)

1 1 hr 1500rpm average 30 Nm load
2 1 hr 2800–3600rpm 80 Nm load
3 ½ hr 3600–4500rpm 100 Nm load
(During the running-in phase the rig varies the load/speed at approx 50-second intervals.)
4 Carry out compression test

HOT-OPEN THROTTLE COMPRESSION TEST (lbf/in²)			
1	2	3	4
205	195	210	210

5 Change to race plugs – ready for load testing
6 Allow engine to cool and retorque head (if applicable)

On a computerized dyno, readings of atmospheric temperature, pressure and humidity are fed automatically into the computer to produce 'corrected' readings. This allows accurate comparisons to be made between different engines on days with varying weather conditions. The difference ordinarily between outputs from the same engine, for example tested on a hot dry day and then a humid cold day, could otherwise be markedly different. Low ambient temperatures and high humidity increase power output (and may affect fuelling requirements).

Test data for power runs can be recorded manually or, if the test rig has the facility, in steps of 250 or 500rpm. If the rig is equipped with a fuel-flow meter, this device will respond more quickly than CO and HC meters, and when linked to a computer printer, a full-power run can be carried out in a matter of minutes, minimizing the stress on the engine.

A Lambda-Sond air-fuel ratio also reacts quickly and is a useful indicator of fuel/air ratio.

The fuelling throughout the range can be optimized from an analysis of the bsfc figures, possibly backed up by the CO/HC, Lambda-Sond readings. After each alteration to the fuel settings (and possibly distributor settings or ignition map) the power run is carried out again, taking care to ensure that coolant and oil temperatures are kept the same.

With a well-equipped rig it is easily possible to run 15–20 load tests per hour in perfect safety! Back-to-back tests of choke size, rampipe design, exhaust manifold and silencer design can also be included and results accurately analyzed.

Ideally, the engine should be tested with the exhaust manifold to be used in the actual car, though due to the design of the dyno rig, this may not always be possible. Where the actual manifold is not used (*eg* engine No 210 had to be tested with a 4-1 for this reason) ideally the fuelling in the vehicle itself should also be checked, either by rolling-road or through track-testing.

[*Author's note:* For a variety of reasons which, strictly speaking, are outside the parameters of this book, GCT have never had much success with rolling-roads. For this reason, the outputs quoted in this book have all been derived (unless otherwise stated) from bench dyno tests. For absolute accuracy, bench dyno tests are essential, although rolling-road results, if corrected for atmospheric conditions, are quite useful for back-to-back tests. Gerard Sauer has kindly produced some useful hints on rolling-roads and their use – *see Appendix D.*]

TYPICAL POWER RUN					
SPEED (rpm)	CORR BRAKE TORQUE (Nm)	CORR BRAKE POWER (kW)	BSFC (gm/kW hr)	OIL TEMP (°C)	COOLANT TEMP (°C)
4000	183.5	76.9	409	89	71
4500	190.9	90.0	388	88	71
5000	190.1	99.5	353	88	71
5500	197.2	113.6	324	88	72
6000	191.9	120.6	340	88	72
6500	187.3	127.5	333	88	72
7000	162.9	133.1	328	87	72
7500	162.9	127.9	349	88	72

CASE HISTORY No 8

Owner Dave Light
Type Lancia St II 2014cc
Use Ferrari replica
Tested Jan '91
Rig Go Power

Specification: GC Eng No 110
Cast pistons, 10:1 CR.
42.5/37 race valves.
IIIA cams timed at 110°.
Ignition timing 35° @ 5000
Fully ported/blueprinted head.
Lightened/balanced flywheel/crank assembly.
Race sump.
Dual interference springs.
45 DCOE (40-choke) carbs.
Clutch Sachs Gp N.

This engine was tested on a dyno rig far less sophisticated than the computerized Superflow. The load had to be applied manually and optimized at each speed to show the best torque result. Mixture checks were made purely on the basis of plug checks at various full-throttle speeds. The engine was run in for three hours (later tests showed that 1½–2 hours was quite adequate) and the following optimized results obtained under load testing.

SPEED (rpm)	TORQUE (corrected lbf ft)	POWER (corrected bhp)
4500	143.5	123.0
5000	149.5	142.4
5500	155.5	162.9
6000	149.3	170.6
6500	143.5	177.7
7000	137.5	183.3
7200	124.6	170.9

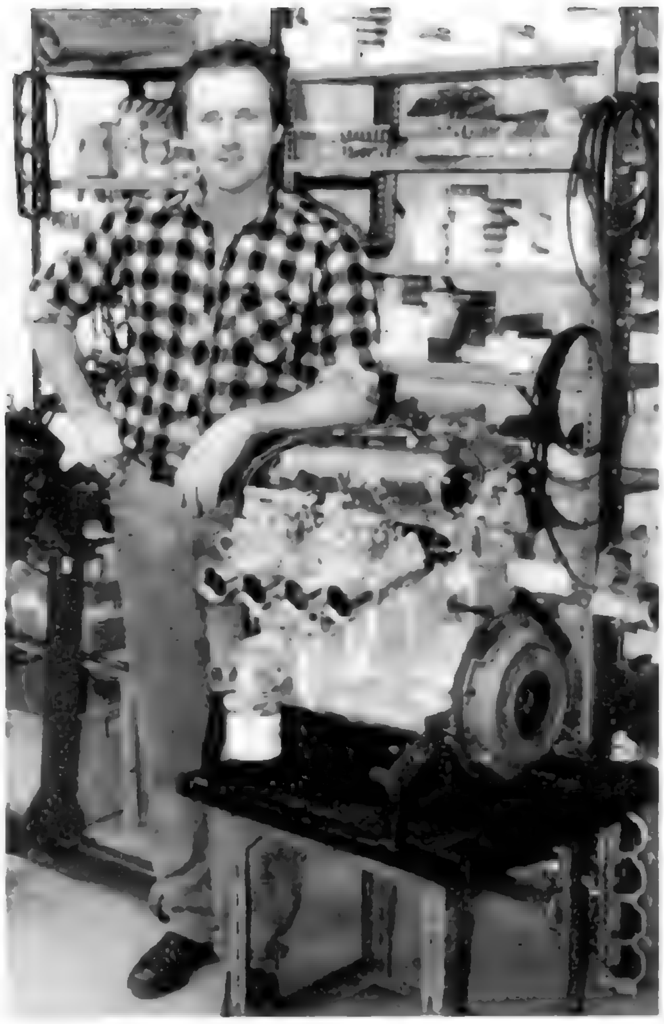
CASE HISTORY No 9

This was a Superflow dyno test of Nigel Peyton's 2*l* Lancia in Aug '91, engine No 88. This was mechanically identical to 110, and the results obtained have been plotted on the same graph. The same 4-2-1 Ansa manifold was used in both tests.

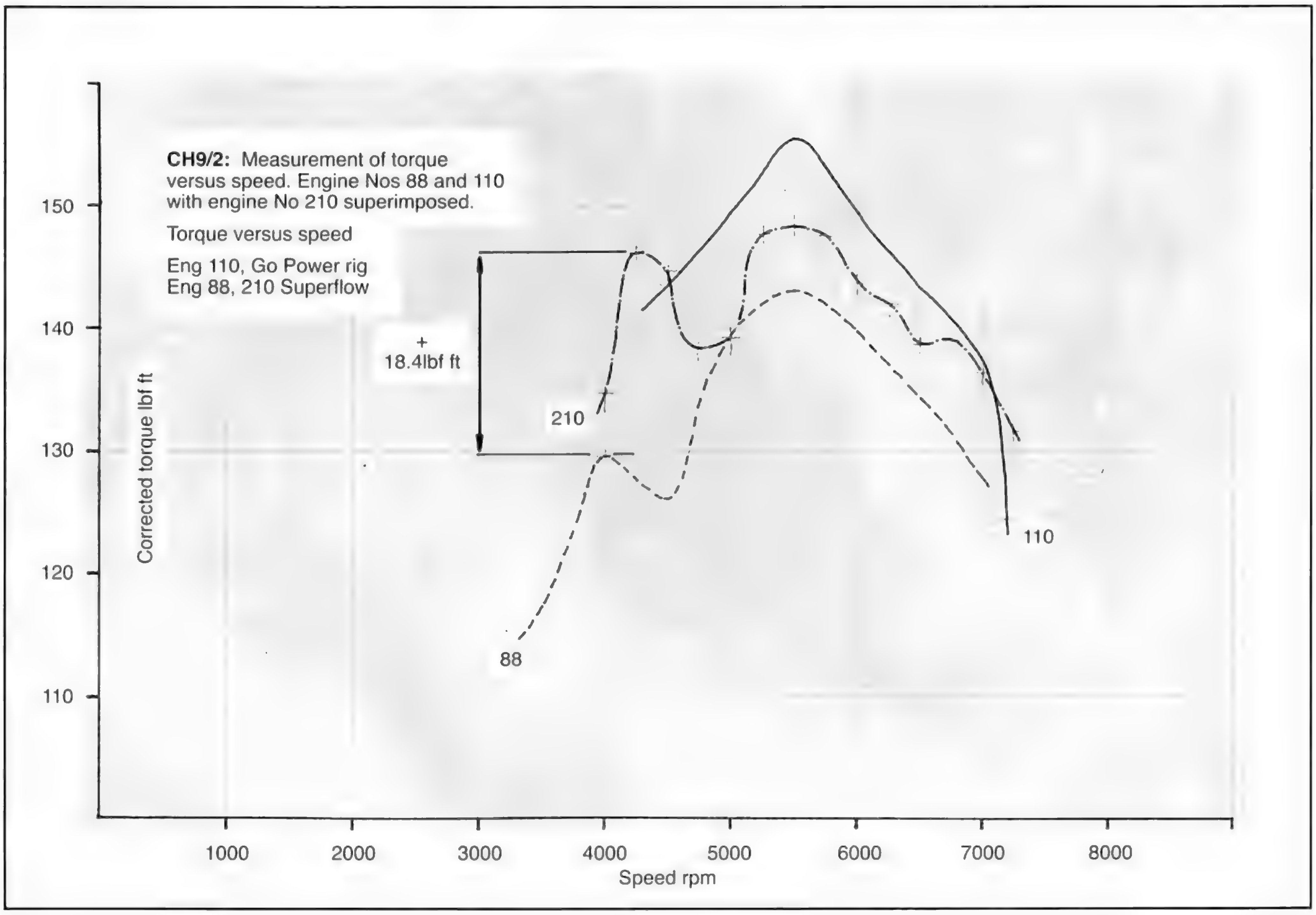
Conclusions: Two identical engines, but two radically different results on two difference dynamometers! A disparity of 12.5lbf ft torque at 5500rpm (even more at 4500). One may reasonably draw the conclusion that the Superflow rig, being vastly more sophisticated (and expensive!), is more accurate, in other words, there may have been an error of 10–14% on the Go Power rig in this case. This is probably due more to the super-sensitive load cell used on the Superflow rig than to any calibration or operator error.

For an interesting comparison, the demo engine results (engine No 210) have been overplotted. This engine has inlet valves 1mm bigger (not race pattern), CR of 11:1 (compared with 10:1), an extra 77cc cubic capacity, but exhaust valves 1mm smaller (standard pattern) and 38 rather than 40mm chokes. Remember, too, this engine was tested *(Case History No 1)* with a technically 'wrong' exhaust manifold, *ie* 4-1 rather than 4-2-1. The significant torque increases from these relatively modest changes are readily apparent on the graph.

Note that the smaller exhaust valve does not inhibit the output in any way. (This is borne out in the flowbench tests in *Chapter 5*.)



CH9/1: Author with Dave Light's 2l Lancia, GC engine No 110. Note Sachs Gp N clutch, essential for engines with around 160bhp/144lbf ft torque.



CH9/2: Measurement of torque versus speed. Engine Nos 88 and 110 with engine No 210 superimposed.

CASE HISTORY No 10

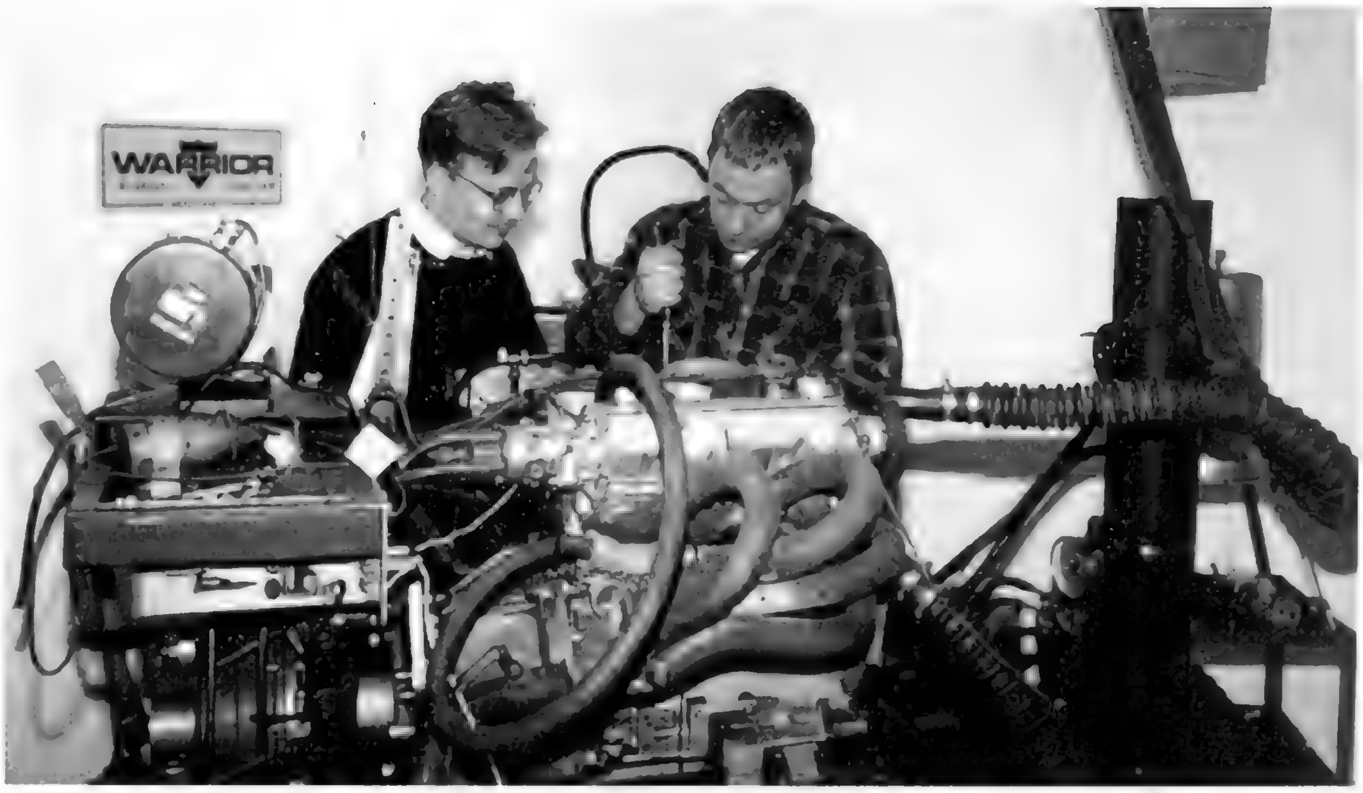
2*l* NHRA Fiat

Tom Casey's Hot Rod engine had been dyno-tested at Warrior Automotive *(see Case History No 2)* with inlet manifold extensions. In order to fit the car, Tom had to shorten them by approx 2" and offset them to the rear of the engine to clear the front nearside suspension. The effect of this was not certain, but would have been expected to alter the position of peak torque and possibly upset the dyno jetting. In point of fact, the results were by no means disappointing, Tom gaining 2nd place in the Ringwood, Hants, European Championship in 1995, 'bumper to bumper' with the two fastest Vauxhall 16v cars.

Tom was convinced that the torque 'out of the corners' was perfect, but the engine lacked power into and at the end of the straights. In order to raise the top-end torque (6000rpm-plus) it was decided to try various new camshaft set-ups, hopefully without upsetting the strong acceleration around 5000rpm. By now it had been determined (with consistently accurate feedback from this top driver) that virtually all racing with the TC took place between 5000–8700rpm. However, it might be that in 'traffic' good torque below 5000 *as well* would be necessary. A tall order!

The exhaust cam was swapped to IIID: no great difference, even with careful trackside jetting. ITG rampipes were also added. A 'hybrid' inlet cam was then designed; this incorporated about the same duration as the IIID but (with oversize bearings) 12mm actual lift. (This had been suggested by Tim Swadkin after the last dyno test.) The base circle and hence LATDC were the same as the IIID and the new lift integral was kept broadly the same, although (owing to the high spring loading) the dwell at FL was reduced. Initially, the results were disappointing – only 3/10 sec improvement at the Rose Green, Tipperary circuit. Only when the IIID exhaust cam was brought forward from 100° BTDC to 105° did the engine 'light up'; this retiming immediately altered the exhaust note, indicating that a much stronger exhaust pulse was present, giving rise to much lower cylinder pressure at TDC as the inlet valve opened.

Although the jetting was believed to be over-rich (and dyno-testing would confirm this) the engine had its most successful outing of '95 at the Foxhall, Ipswich Stadium in the autumn. A 'shunt' at the start of Tom's first heat put him off with a damaged rear arch (rubbing on the tyre and smoking – like a 'blown' engine!). In his next heat, Tom started 4th alongside his Irish friend and rival Neville Stanley (SHP car – Mass Peugeot 2*l* engine). In a ferocious standing-start, Tom shot past Stanley immediately the flag dropped, caught and took the two front cars (Mass engines) and started driving away from the pack. He finished 1st after 25 laps by a full straight. In the Championship race, Tom was seeded 15th and fought his way through the pack (skilfully avoiding numerous pile-ups) to finish 4th at the end of the 65-lap race: overall 5th. Tom's verdict – the quickest engine yet, but still short of some 'top-end'. (By now other racers were starting to take the Fiat seriously – the main rivals being the Mass unit and Vauxhall 16v – albeit limited to 38mm chokes and 8000rpm.) Some 'top-end' was being lost due to over-rich mixture *(Graph 1)*. 'Back to the drawing-board' it was decided to dyno the engine 'as is' and then optimize the set-up as the mixture and balance were by now way out! In preparation for the dyno test, the VP19 rod bearings were replaced (the old ones were in quite good shape), a compression test was carried out, and two carburettor O-rings replaced – the old ones had virtually disintegrated – and the cam belt was changed. The cam timing and lift were carefully assessed, as follows:



CH10/1: Tom Casey's engine on the dyno at Warrior. Engineer Mike Hockley swaps main jets to lean-out the mid/top-end mixture on the 48 DCOEs between power runs. Simon McCann (left) started training at GCT in 1995 during time-off from his college BTEC Automobile Engineering course. The distributor for Tom's engine is a composite, made from a Saab body with Vauxhall SOHC intervals and power pack.

Hybrid inlet cam	LATDC 4.7mm (clearance 6 thou")
	Lift 12mm actual
	FL at 107° ATDC
	Duration 333°
IIID ex cam	LATDC 4.5mm (clearance 8 thou")
	Lift 11.1mm actual
	FL at 105° BTDC
	Duration 305°

(*Note:* Valve clearances tighter than standard, leading to longer duration.)

On Warrior's dyno (Superflow rig – **CH10/1**) the results of the engine 'as-is' were assessed against it with optimized jetting and balance *(see Graph 1)* and the earlier engine used in the European Championship at Ringwood, Hants.

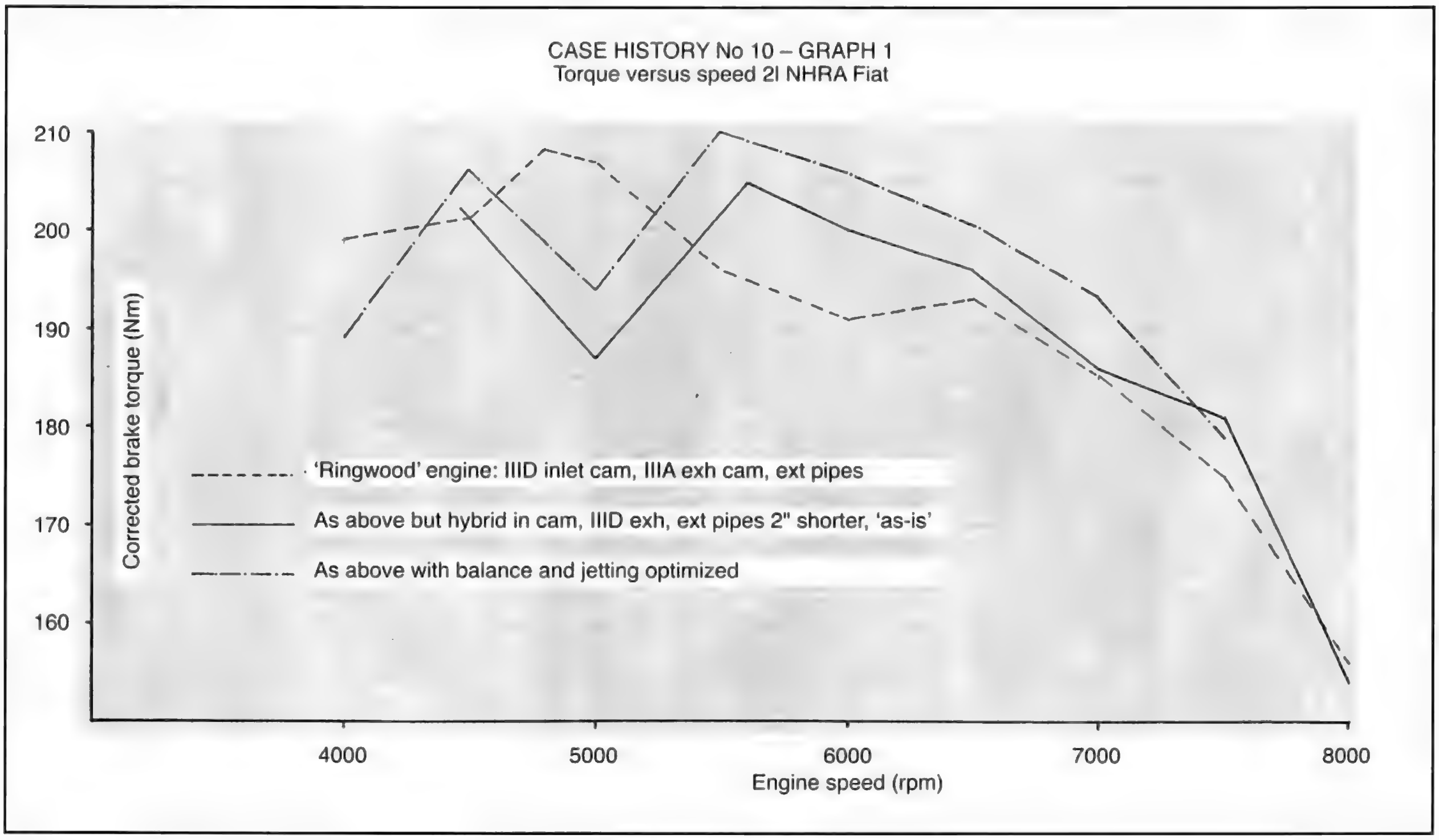
Ignition advance was 18° @ 1000rpm
38° @ 5000rpm

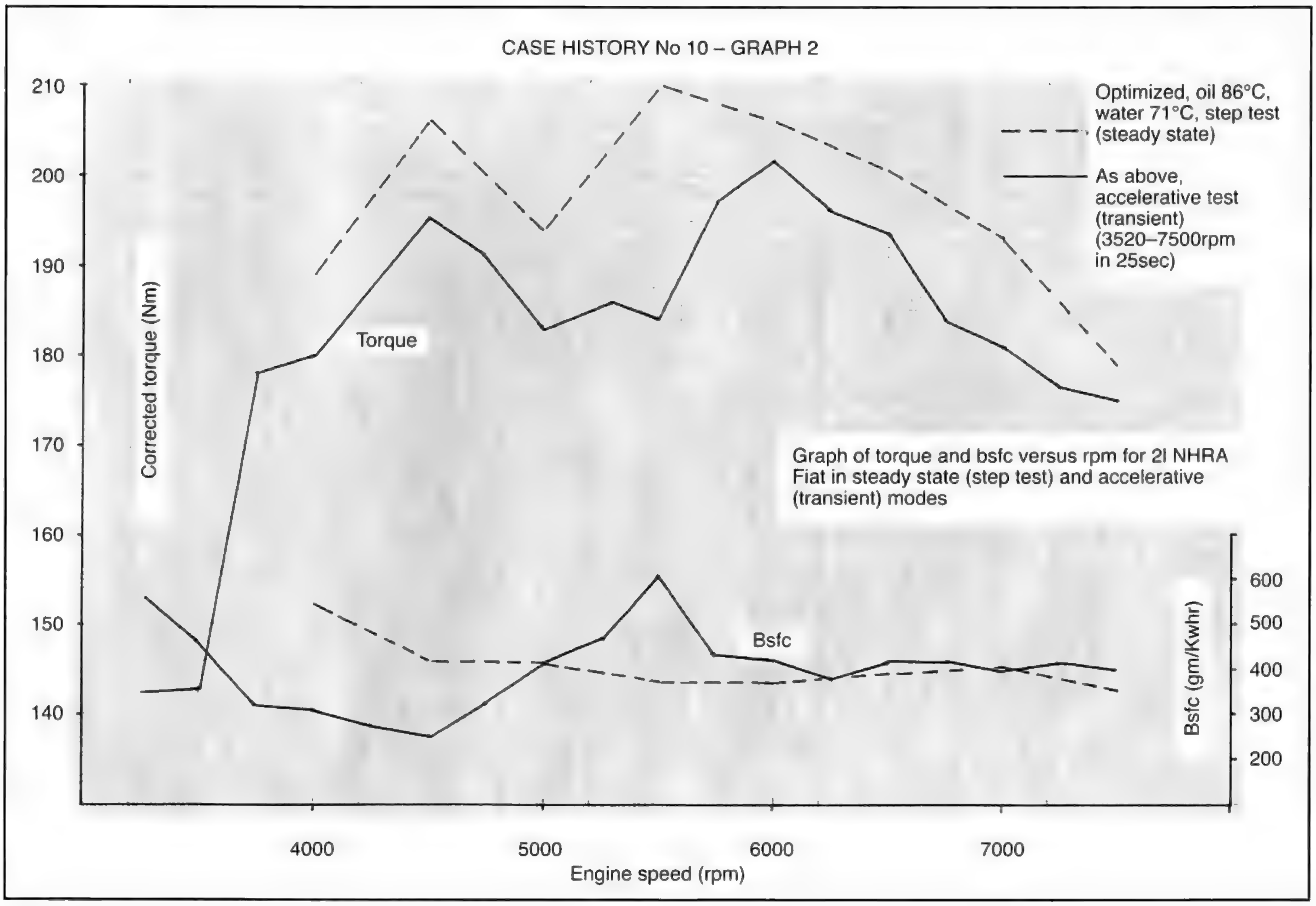
Clearly, adoption of the new cam set-up and rampipes had achieved the desired result – compare the optimized engine with the 'Ringwood' engine. The new engine had significantly greater torque from about 5200rpm upwards, and about the same around 4500. Particularly, the 'hole' in the torque curve at 5500–6000rpm had been turned into a major gain. (The engine was not taken over 7500, to avoid over-stressing what was essentially a 'tired' unit, as it was intended to use it for back-to-back tests on the same session.) The hot, cranking compressions (lbf/in^2) were:

1	2	3	4
185	200	205	180†

†(±12% – not bad after a hard year's racing with incorrect jetting)

(Ring gaps on strip-down were exactly as built, at 15 thou", although there was some inlet valve seat wear.) The jetting


CASE HISTORY No 10 – GRAPH 1
Torque versus speed 2l NHRA Fiat
Corrected brake torque (Nm)
210
200
190
180
170
160
'Ringwood' engine: IIID inlet cam, IIIA exh cam, ext pipes
As above but hybrid in cam, IIID exh, ext pipes 2" shorter, 'as-is'
As above with balance and jetting optimized
4000
5000
6000
7000
8000
Engine speed (rpm)


CASE HISTORY No 10 – GRAPH 2
Corrected torque (Nm)
210
200
190
180
170
160
150
140
Optimized, oil 86°C, water 71°C, step test (steady state)
As above, accelerative test (transient) (3520–7500rpm in 25sec)
Torque
Graph of torque and bsfc versus rpm for 2l NHRA Fiat in steady state (step test) and accelerative (transient) modes
Bsfc
Bsfc (gm/Kwhr)
600
500
400
300
200
4000
5000
6000
7000
Engine speed (rpm)

was progressively leaned-out from 175 main to 165 main, in the process picking up:

4.7lbf ft @ 6000rpm
0.9lbf ft @ 6500rpm
4.22lbf ft @ 7000rpm

Peak torque was 209.9 Nm (corr) @ 5500rpm (155.3 lbf ft) with:

@ 7000 rpm 189 bhp (corr)
@ 7500 rpm 188 bhp (corr)

(cell temp 13°C – negative correction)
These results were obtained with oil temp 84°C, coolant 71°C.

Transient response

This test was to establish the torque characteristic of the engine under acceleration. Most dyno tests hold the load constant for a period of some 5sec or more in order for the airflow/fuelling conditions in the inlet tract to stabilize. In this particular test, the load was reduced progressively over a 25sec interval and readings of torque and bsfc taken at 250rpm intervals. By unloading the engine in this way (at full-throttle) the transient response of the engine could be examined. The results are both useful and interesting since they give an indication of the throttle response of the engine. This response is a function of inlet tract pressure wave effects, column inertia and carburettor signal. The results for 'steady-state' torque (from Graph 1) are plotted against the accelerative condition at Graph 2.

There are a number of salient points from the graph which are worth highlighting.

Torque-shift – the transient torque peaks at higher rpm than the steady state. Bear in mind that the test period was 25sec and in fact in NHRA this time frame is compressed into the time for the vehicle to accelerate from the turn exit (at 5000rpm) to the end of the straight (around 8000–8600rpm). Thus, as peak torque comes in higher, provided the initial torque to leave the turn is adequate, this characteristic is probably an advantage as, by the 6050rpm point the vehicle is safely on the straight, where maximum acceleration can be used. The average transient torque is measurably weaker than the steady state, but at least it is building up at the right place in the rev-band. It may be assumed that the harder the engine is accelerated, the greater this torque-shift tendency will be and, for optimum performance, it should be taken into account at the engine design stage.

Fuelling – below 5000rpm the transient test demonstrates a tendency for the engine to run weak, whereas around 5500rpm overfuelling occurs. These effects are mirrored in the torque response, and the best transient torque is obtained above 6000rpm, where the fuelling is broadly similar, in terms of the bsti, to the steady-state test. Clearly, around 5500rpm the transient unit struggles to inhale enough air; possibly the pressure-wave generated in the inlet tract is related to piston speed and at 5500 is not travelling fast enough to ram-charge the cylinder before the inlet valve closes. The results improve at higher speeds.

Note that, on the flexibility test, this engine sustained full-throttle, with maximum load, down to an amazing 2500rpm.

CHAPTER 17

CLUTCHES AND GEARBOXES

Clutch designs

The clutch is required to transmit the torque from the engine to the gearbox. It is called upon to do this repeatedly, under conditions of heat and stress, and the marriage of a suitable design to the engine is a factor often overlooked by owners, who, having built an engine of a given bhp/torque, may baulk at the last minute at the idea of having to spend a not inconsequential sum on a clutch which is up to the job!

As a general guide, manufacturers' standard clutches will cope with about 15% extra torque under steady load – and this should not be taken to mean that they will withstand standing starts/race use. **(17/1, 17/2)**

The torque transmission of which a clutch is capable is essentially derived from the following equation:

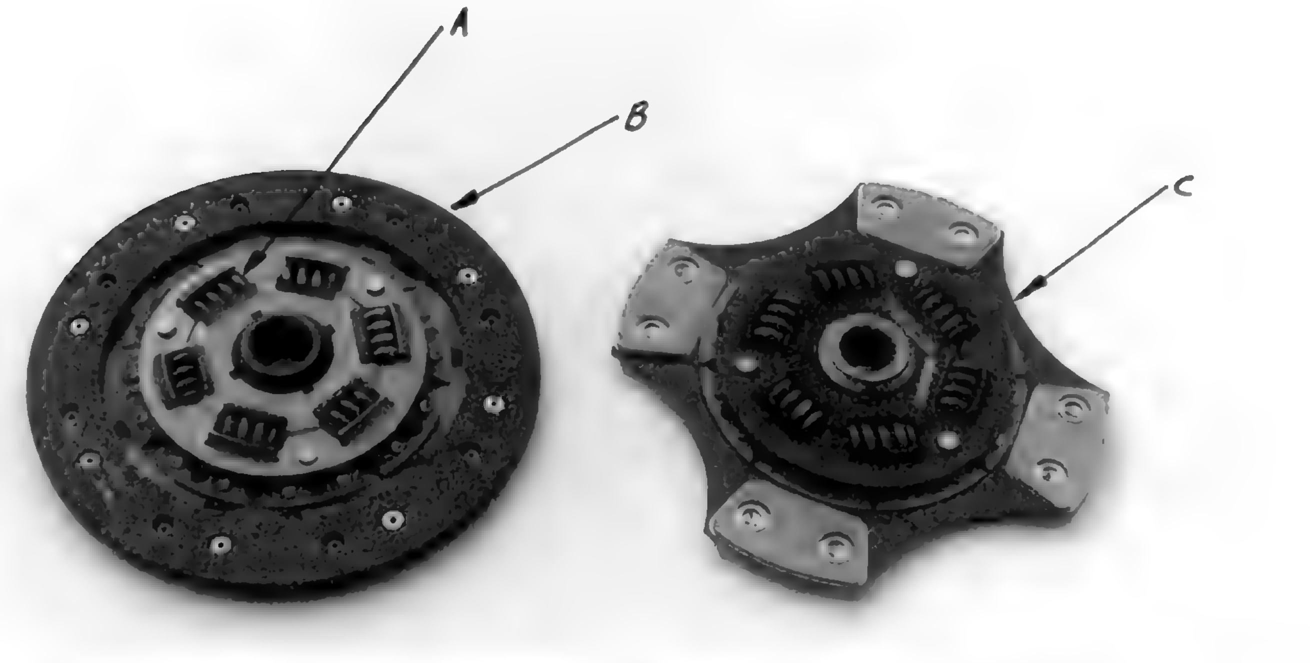
$$T = F\,R$$

where $F = \mu W$
and T = torque transmitted (Nm)
F = friction force (N)
R = clutch radius (m)
μ = coefficient of friction
W = clamp load (N)

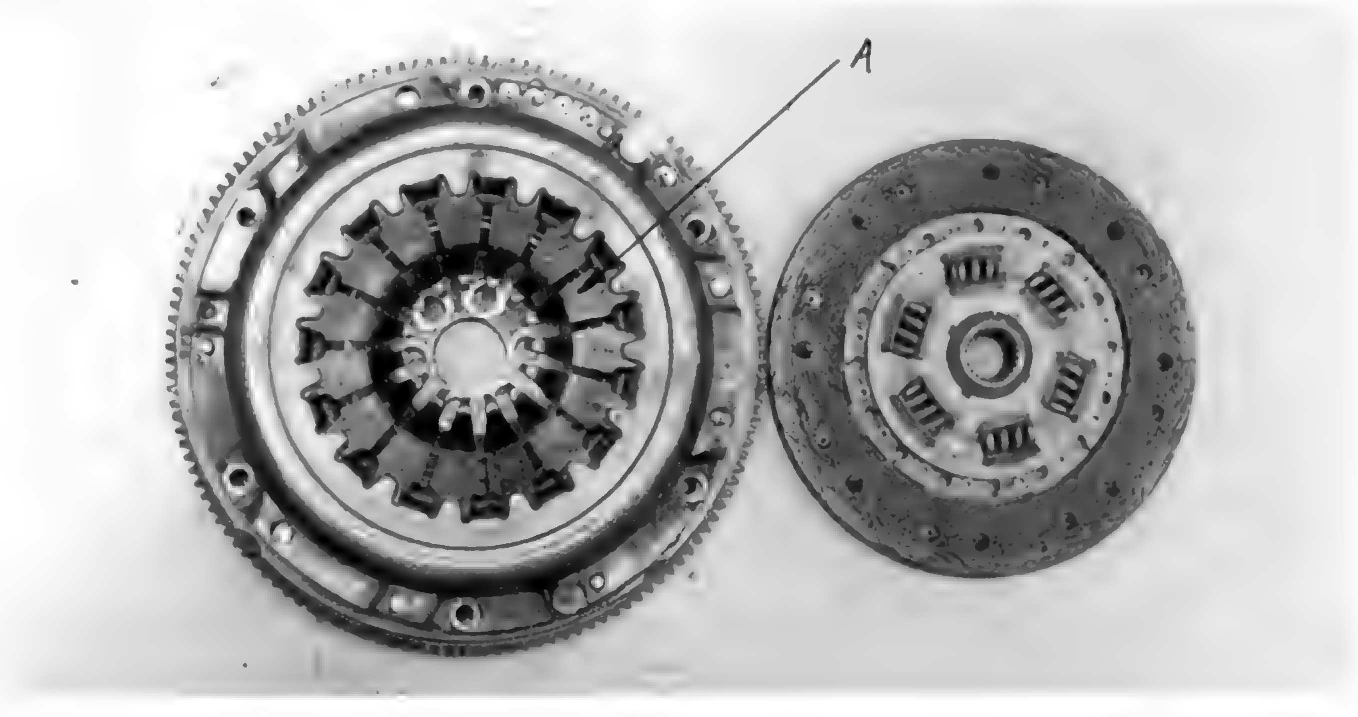
(For a twin-plate clutch, $T = 2 \times FR$). Hence it may be seen that for a given ⌀ of clutch plate (131 1600 – 190mm, 2l 215mm) the transmissible torque depends on the coefficient of friction and clamp load.

Clearly, the clutch manufacturer must ensure that the rest of the construction of the unit is sufficiently robust to withstand the forces generated, *ie* strong rivets (and bonding adhesive in the case of competition clutches) to hold the facings in place on the backplate, a backplate of sufficient thickness to withstand the load from the torsion springs, strong driving straps on the cover to transmit the frictional force between the contact faces without distortion, and so on.

[*Author's note:* These are the friction plate diameters, not the flywheel face diameters – generally 5mm larger.]



17/1: Left: GCT Kevlar plate (1" × 23-spline) to marry 131 2l flywheel/cover to Ford box. Torsion springs (heavily uprated) are at A. Heavy-duty facing B is bonded and rivetted to backplate. With Sachs 2l Gp N cover – will accept 160lbf ft torque. Right: Spring cerametallic 'paddle' clutch plate will cope with 30% more torque than Kevlar or organic item. This one is for a Volumex: 131 2l flywheel, Gp N cover. Backplate is at C.

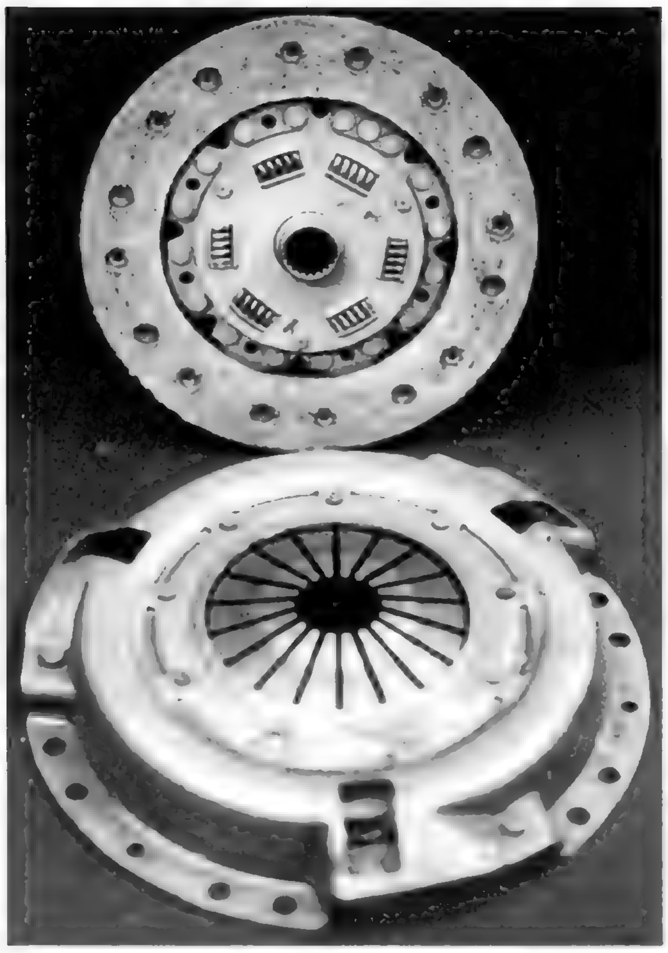


17/2: Conversion flywheel/clutch 131 2l – Ford Sierra 5-speed box. Cover/plate is 1" × 23-spline AP Racing RS 2000. Cover springs are at A. Combination good for 150lbf ft torque.

Clutch failure and selection

The first, and easiest, way to damage a clutch is during fitting; never allow the gearbox to 'hang' on the clutch. This causes the torsion springs to be pulled out of their housings and distorts the backplate.

It is common practice these days for manufacturers to specify that the release bearing should run in light contact with the cover springs. Early Fiat/Lancia TCs ran with about ⅛" free play on the clutch actuating lever (at the cable end). Either method is acceptable, but excessive pressure on the springs leads to wear and may allow the plate to slip. This leads to facing and flywheel damage caused by overheating.



17/3: Sachs Gp N clutch for 1600 131 will accept around 140lbf ft torque. Only available with ⅞" × 20-spline, but other types can be made up. Clutch type is sprung, uprated organic. Heavier torsion springs, stronger facings, plus facings bonded and rivetted to backplate. Clamp load of cover is also heavier. Standard of balance of Sachs clutches is excellent – no need to balance with flywheel. 2l version is similar, but larger diameter. Conventional cover design can also be used with spring (or rigid) cerametallic or sintered paddle plate for turbo engines where organic plate will not withstand torque and heat.

The clutch must be matched to the torque output of the engine and have facings (organic/Kevlar or sintered/cerametallic) capable of withstanding the type of use to which the car will be put. If vicious standing starts with a high-powered engine are likely to be encountered, *eg* NHRA, the enormous energy fed through the clutch may require a clutch capable of withstanding two or three times the 'straight line' engine torque rating. (17/3, 17/4)

Organic facings will tolerate repeated slip, but tend to be bulky. Therefore, for high rpm the smaller race 7¼" or 5½" design is preferred. For very high torque applications, it is necessary to go down one of three routes:

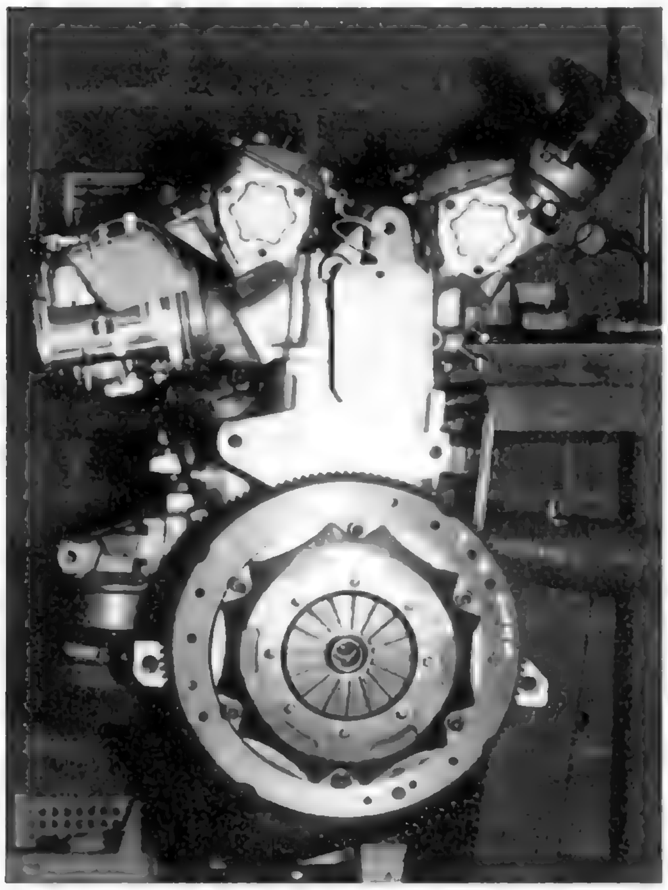
1 Larger diameter (8v Integrale clutch will take over 225lbf ft torque).
2 Combination of conventional, standard dia/high clamp load cover mated to cerametallic or sintered paddle-type friction plate. These facings have 20%-plus more friction than organic/Kevlar.
3 7¼" single or multi-plate design (these clutches are available with paddle or disc plates, according to the durability required). (17/5)

Use of an inadequate clutch may cause the facings to break up, leading to loss of drive and possible cover/flywheel damage. Because of the complication of stripping out the assembly, it is far better to choose the right clutch 'first time', integrating this selection into the overall specification of the powerplant at the initial budget stage. If the engine is modestly tuned at first, but a cam swap is envisaged at a later date, fit a heavier clutch from day one! Ultimately, the choice of clutch depends on the type of use, torque, rpm and type of gearset.

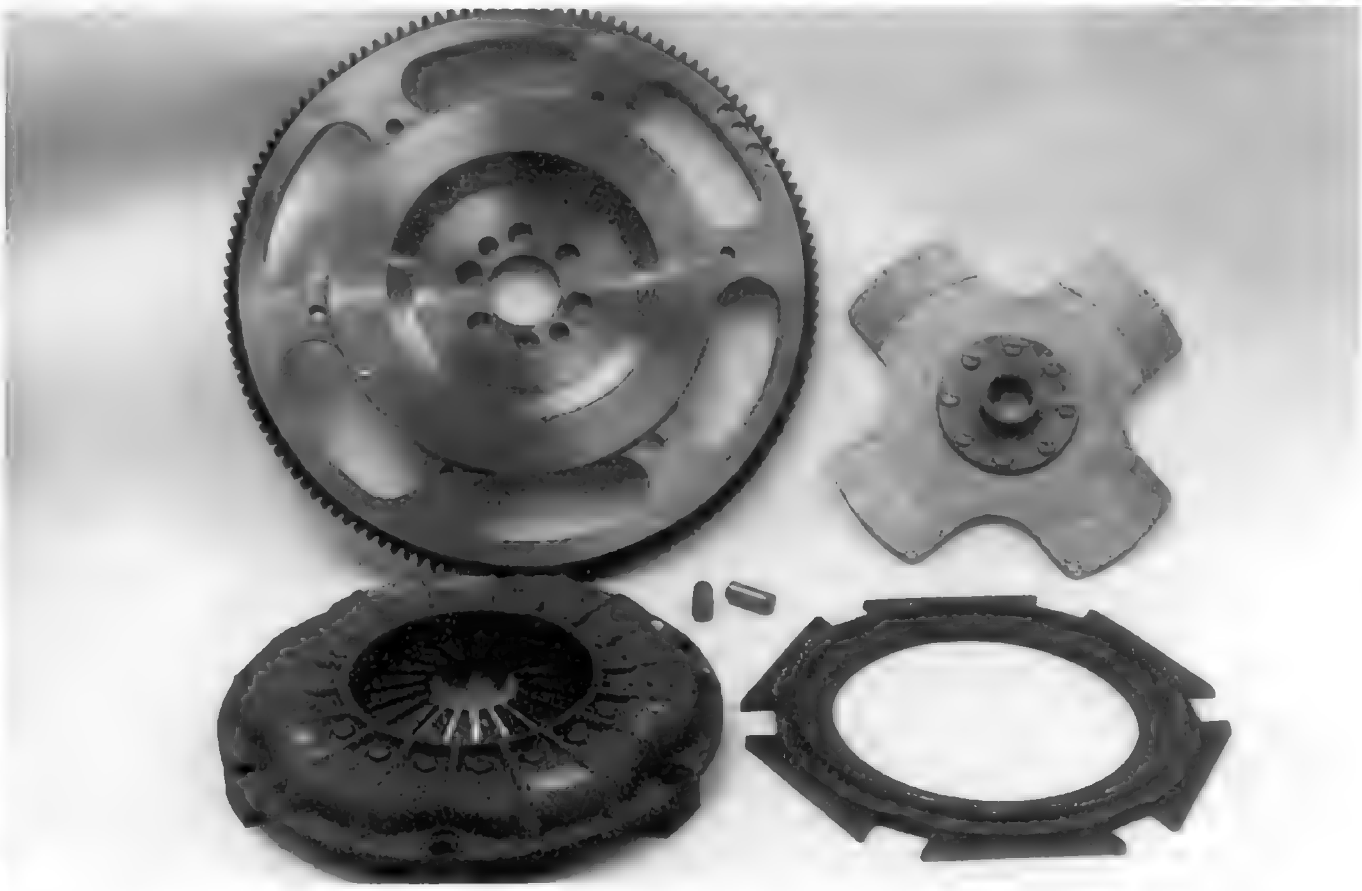
Suggested clutch choices:
Road use and road/race:

A	Up to 15% extra torque	–	uprated organic plate/std cover
B	Up to 30% "	–	uprated organic plate/uprated cover
C	Up to 60% "	–	uprated cover/cerametallic (paddle)
D	Up to 50% "	–	larger diameter clutch with combination of above.

Examples:
– GC-tuned 1.6 Delta turbo *ie*, 220lbf ft torque on 20lbf/in^2 boost – standard flywheel with combination (C) above (56% extra) (GC clutch)
– GC-tuned 131 2*l*, 142lbf ft torque – (15% extra), std clutch (AP preferred)
– GC St II 131 – 2.1*l*, 148lbf ft torque (20% extra) Sachs Gp N cover/plate



17/4: Tilton 7¼" race clutch (single-plate) fitted to Monte Carlo race engine. Flywheel is modified, lightened cast iron – OK for competition use up to around 7000rpm, but crack test each season, especially around centre. This particular design of clutch suffered stress cracks between plate rotors due to inadequate radius.



17/5: Components of Sachs 7¼" sintered clutch with ultra-light steel flyweel. Flywheel has been balanced with crank (dowels fitted). Sachs race clutches give exceptional performance – this lightweight model will easily last a season's circuit racing on a St IV engine. Note generous radius between rotors of clutch disc; 7¼" race clutches can be built up into double or even triple-plate types for high-torque applications. Single-plate shown will easily transmit over 300lbf ft torque. Main advantages of these types of clutches are torque capacity and light weight. By reducing diameter of clutch, inertia is reduced, leading to quicker throttle response. A wide variety of different spline sizes are available.

Pure competition:
As road use, but with the following additional comments:
In view of the high rpm, use the smallest clutch available – for example, on the 1585 use a Sachs Gp N or equivalent rather than a 2*l* flywheel/clutch combination. Consider the use of a 7¼" diameter clutch, especially for circuit, sprint/hillclimb and oval racing, and particularly when using a close-ratio gearbox, as this type of clutch gives a significantly swifter gearchange. Remember that 7¼" clutches will not tolerate slip (hill starts, loading car onto trailer, etc) and therefore for rally use in particular it may be worth using a twin-plate 7¼" as a means of extending the life of the clutch – even though the torque capacity of a single-plate item may be adequate.

Turbocharged 2*l* rally cars, because of the savage torque response 'on boost', require the power to be fed in gradually for the first few yards of car movement to avoid drivetrain damage. 'Dumping' the clutch at high rpm is definitely out! The torque 'off boost' may be quite poor, and the clutch may need to be feathered to get the car moving whilst feeding in the power gradually. For this reason, a multi-plate 7¼" clutch may not be the best option.

Cars for oval and grasstrack, where no gearchanging is required once the vehicle has left the startline, are best equipped with a 7¼" clutch.
Examples:
GC St III rally 1600 Delta, 125lbf ft torque (27% extra), (peak 9500rpm) – steel flywheel, twin-plate 7¼" (17/6).
GC St III National Hot Rod 2*l* Fiat, 154lbf ft torque (25% extra), (peak 8500rpm) – steel flywheel, twin-plate 7¼".
GC St III rally 2*l* 131, 152lbf ft torque, (peak 7200rpm) – standard flywheel/Sachs Gp N (uprated organic cover/plate).
GC road/race Volumex (Stratos replica), (180lbf ft torque (18% extra), (peak 7000rpm) – standard flywheel, GC-design sprung sintered paddle/Sachs Gp N cover.
GC Lancia Integrale 8v Gp A rally, 300lbf ft torque (35% extra), (peak 7000rpm) – standard flywheel, GC-design sprung sintered paddle/uprated cover.

Replace the 7¼" clutch plate(s) when the thickness falls more than 20thou" from new (single-plate) and 16thou" (twin).

Release bearings for 7¼": GCT normally use a production type with an En8 insert pressed into the bearing face, increasing the diameter to match the 7¼" spring layout. (This has been used on Sachs twin-plate types with no problems.)

TORQUE RATINGS (lbf ft)

SACHS				
Single-plate, 047 cover				
Sintered	310			
Twin-plate, 048 cover				
Sintered	620			
AP RACING				
SPRING COLOUR	GREEN	BUFF	ORANGE	GREY
Single-plate				
Sintered	144	151	207	277
Cerametallic	109	116	158	210
Twin-plate				
Sintered	258	273	372	498
Cerametallic	197	229	295	368

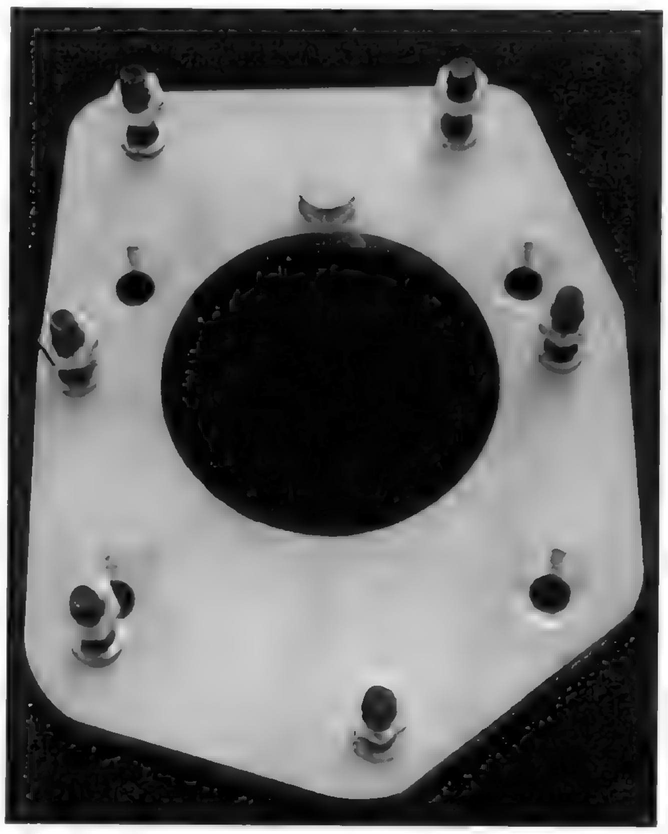
Note: Cerametallic facings are fitted to paddle-type clutches, sintered are disc-type.

Clutch conversions

One of the most popular is the Fiat-Ford Sierra 5-speed box. This gearbox has the larger 1" × 23 spline, compared with the ⅞" × 20 type used on the 131 5-speed box. The easiest way to carry out this conversion is using the AP Racing RS2000 cover and plate *(see photo)*. This requires the flywheel to be machined flat (step removed) and then the dowel and bolt holes redrilled. The modified flywheel *must* then be balanced as follows: Balance crank, balance flywheel, balance flywheel with cover bolted in plate, balance assembly fitted to crank. The Ford release bearing is used. (17/7)

An adaptor plate allows the Ford box to be bolted to the Fiat 124 1800 or 131 5-speed bellhousing. The release bearing actuating arm is made by welding together the inner and outer sections of the Fiat arm with the Ford centre section to carry the bearing. Provided that the adaptor plate is sufficiently thin, the Ford input shaft will locate correctly in the Fiat crank-end bearing. It is possible *(see photo)* to produce a special clutch which bolts directly to the Fiat flywheel to accept the Ford input shaft, eliminating the need to machine the flywheel, but

17/6: 1600 Delta owned by Rod Bennett (164bhp @ 8000rpm) chewed driveshafts like toffee on its first couple of rallies – later changed for turbo items. Combination of twin-plate race clutch and race gearset created massive fluctuation of torsional loading which led to classic shear failure.



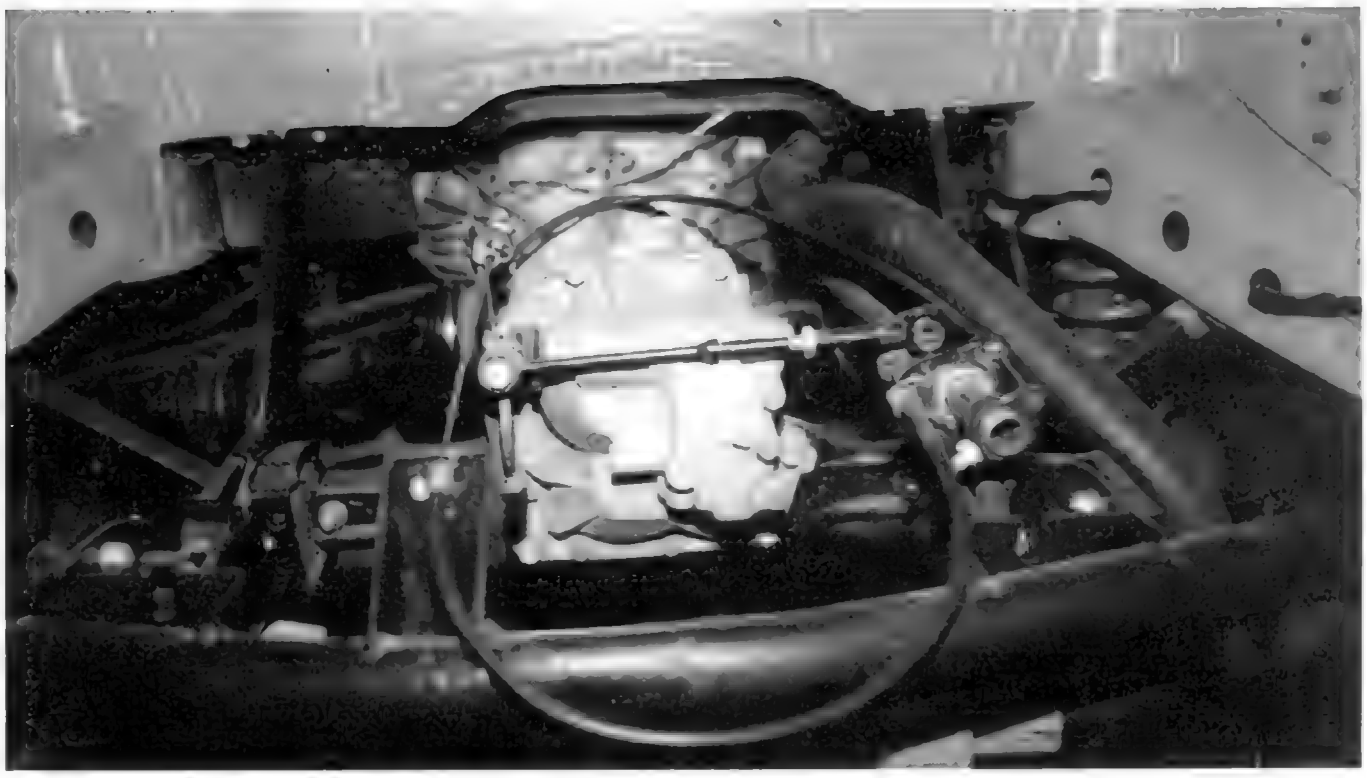
17/7: GC adaptor plate, Ford 5-speed Sierra – 131 2l (also 131 1600 TC, 132 2l) bellhousing. Plate bolts between bellhousing and box, thus standard starter motor can be retained. View looking towards gearbox.

clearly the cover must be designed so that springs engage properly with the Ford release bearing.

A number of owners have used transaxle Renault gearboxes with success (17/8, 17/9). The exact conversions vary between installations, but the GC conversion for the Renault 2*l* turbo box for Midtec sportscars was particularly easy. The 2*l* Fiat flywheel fitted inside the adaptor and bellhousing, so all that was needed was a special design of cover/plate which bolted direct to the standard flywheel and accepted the Renault release bearing.

Many owners use production gearboxes (and gearsets) with great success, but for optimum performance, the gearset should match the output from the engine, in other words, for a given event (race/rally,

17/8: GC CNC machined adaptor allows Renault 2l Turbo bellhousing and box to be bolted directly to back of 84mm bore-type Fiat TC. Adaptor carries standard Fiat starter motor. Clutch from GCT bolts direct to standard pattern Fiat flywheel and accepts Renault spline.



17/9: Renault 25 transaxle gearbox fitted to 131 2l engine in Skoda. Nicely executed conversion by Eric Cox.

etc) the gearset should be so designed as to keep the engine between peak torque and maximum power at all times when underway. Unfortunately, most production gearsets (not just Fiat/Lancia) tend to be designed on the basis of giving maximum driveability rather than maximum acceleration! For maximum acceleration, the power drop between gears should be kept to a minimum. Clearly, as there is a limit to how many gears can be fitted into the gearbox casing, to achieve a high top speed a tall first gear will also be needed – which can present problems getting off the line if the torque is weak at low revs.

The availability of rear-wheel-drive gearsets for the 124 and 131-type boxes is currently zero [special manufacture would be prohibitively expensive], but sets for the Delta, 130 TC and Integrale types are still available in Italy. Optimum performance from a RWD car can also be obtained by conversion to a Sierra Type N 5-speed or RS 2000 4-speed box with Trans-X gearsets.

As an illustration of how effective such a conversion can be, the author would readily trade 20bhp for a close-ratio gearset on a St II 2l! This is a point well worth considering when designing the specification of the car (and working out a budget). The criterion for gearset selection is again the type of circuit or event to be encountered. Oval calls for racing in second or third gear and demands an rpm range of 5000–8500, depending on the final-drive. Hillclimbs may require the car to reach 70mph in second gear (high torque/rpm cam selection or very high torque and a tall differential required). Rally use demands a flexible gearset that will give the car useful torque from 20–110mph (forest) or 40–140mph (tarmac). **(17/10)**

Gearboxes/final-drives

With the notable exceptions of some Delta Turbo *ie* variants, 124 Sport and 130 TC gearboxes, all the Fiat/Lancia units are extremely strong. GCT have used Lancia Beta and 131/132 5-speed boxes with well over twice their design power output.

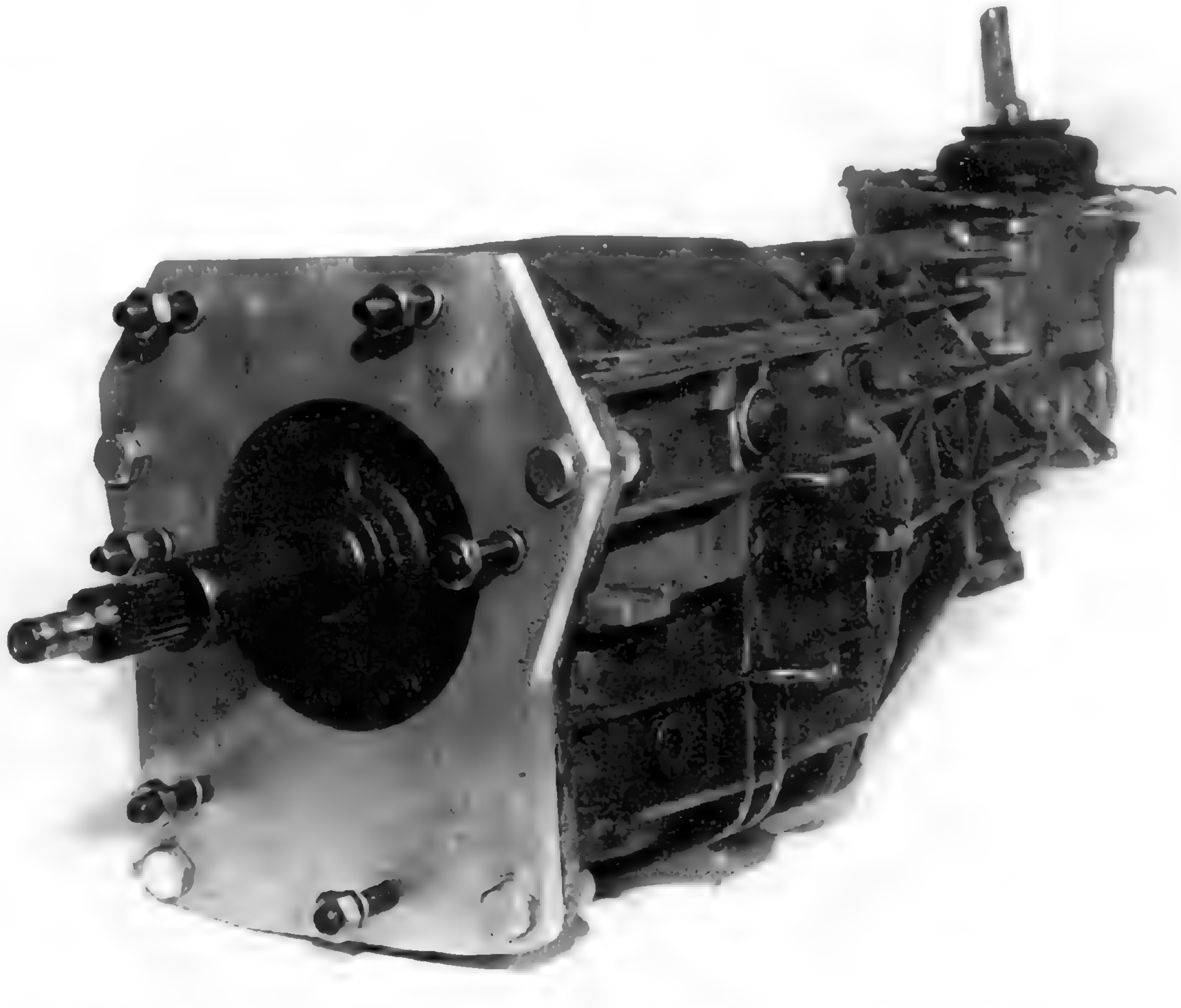
The late rear-drive 5-speed box **(17/11)** is especially popular (particularly if from the 2l 131 Sport two-door, also known as the 131 Racing, with its Abarth-design remote gearchange – **17/12**) for its well spaced ratios, availability and great strength. **(17/13)** Some typical ratios are as follows:

124 1800 Abarth Rally, 124 BC 1608 (ribbed casing)

1st	3.667:1
2nd	2.100:1
3rd	1.361:1
4th	1.000:1
5th	0.881:1

132 1800 GL/GLS (ribbed casing)

	(Pre-75)	(Post-75)
1st	3.544:1	3.612:1
2nd	2.175:1	2.045:1
3rd	1.410:1	1.357:1
4th	1.000:1	1.000:1
5th	0.913:1	0.870:1



17/10: GC adaptor plate bolts 131 2l bellhousing to Cossie T5 5-speed gearbox.

131 Sport, 132 2*l*	
1st	3.612:1
2nd	2.045:1
3rd	1.357:1
4th	1.000:1
5th	0.870:1

(There are minor variations on four-door 2*l* 131 models on 2nd, 3rd and 4th gears, but these boxes to all intents may be treated as being identical.)

125 / 130 TC		105 TC
1st	3.583:1	3.583:1
2nd	2.235:1	2.235:1
3rd	1.542:1	1.550:1
4th	1.154:1	1.163:1
5th	0.967:1	0.959:1

Lancia Beta (all)	
1st	3.50:1 (early)
	3.75:1 (late)
2nd	2.23:1
3rd	1.52:1
4th	1.15:1
5th	0.925:1

Whereas a production gearset might typically be designed so that the power may be allowed to drop in any gear to 65% of maximum, with a close-ratio gearset the gears are spaced such that the power drop between gears is reduced and the rpm drop is kept typically between 1500 (rally) and 1000 (race). For example, if maximum power is at 7500rpm, then minimum rpm (with the exception of 1st gear) will be maintained at 6500, and deceleration due to power loss on the upward gearchange will be minimized. If the ratio of N(rpm) max/N min is defined as r, and the gear ratio in top as R, a true close-ratio gearbox (with the gears in geometric progression and equal power drops) will have gear spacings as follows:

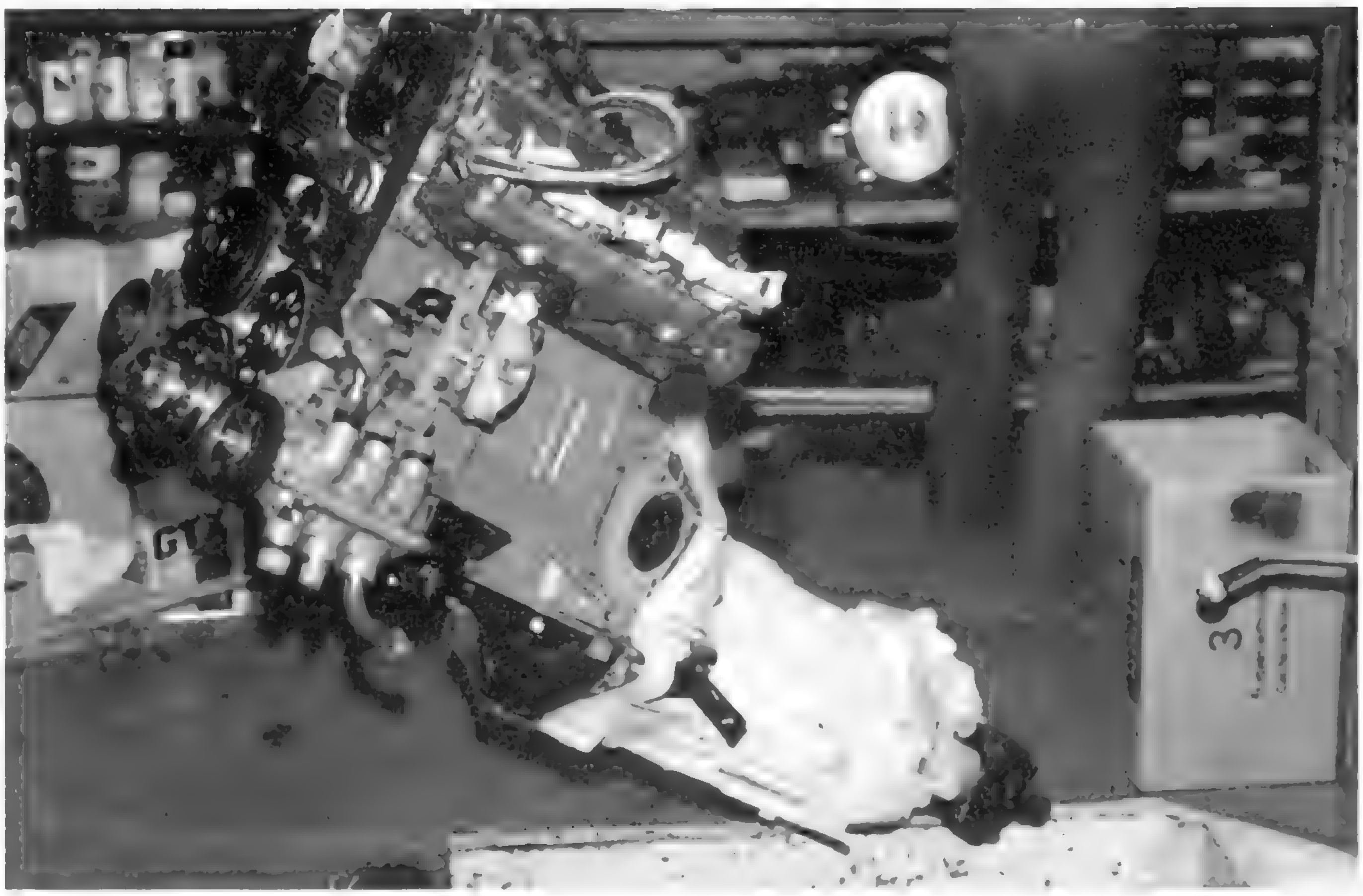
5th	R
4th	rR
3rd	r^2R
2nd	r^3R
1st	r^4R

Using the above (race) example:

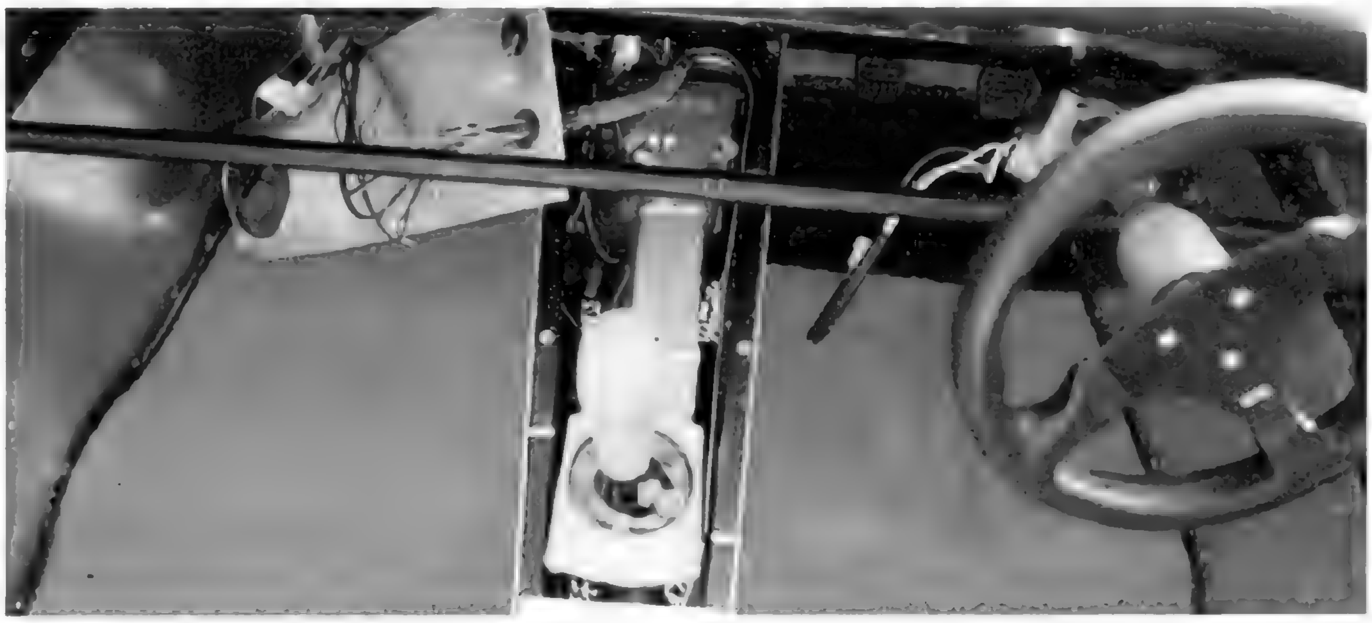
$$\frac{\text{Nmax}}{\text{Nmin}} = \frac{7500}{6500} = 1.15$$

To achieve a satisfactory top speed in top gear, for a given tyre size and differential ratio, R might be set at 0.87. For this engine, the gearsets would then be:

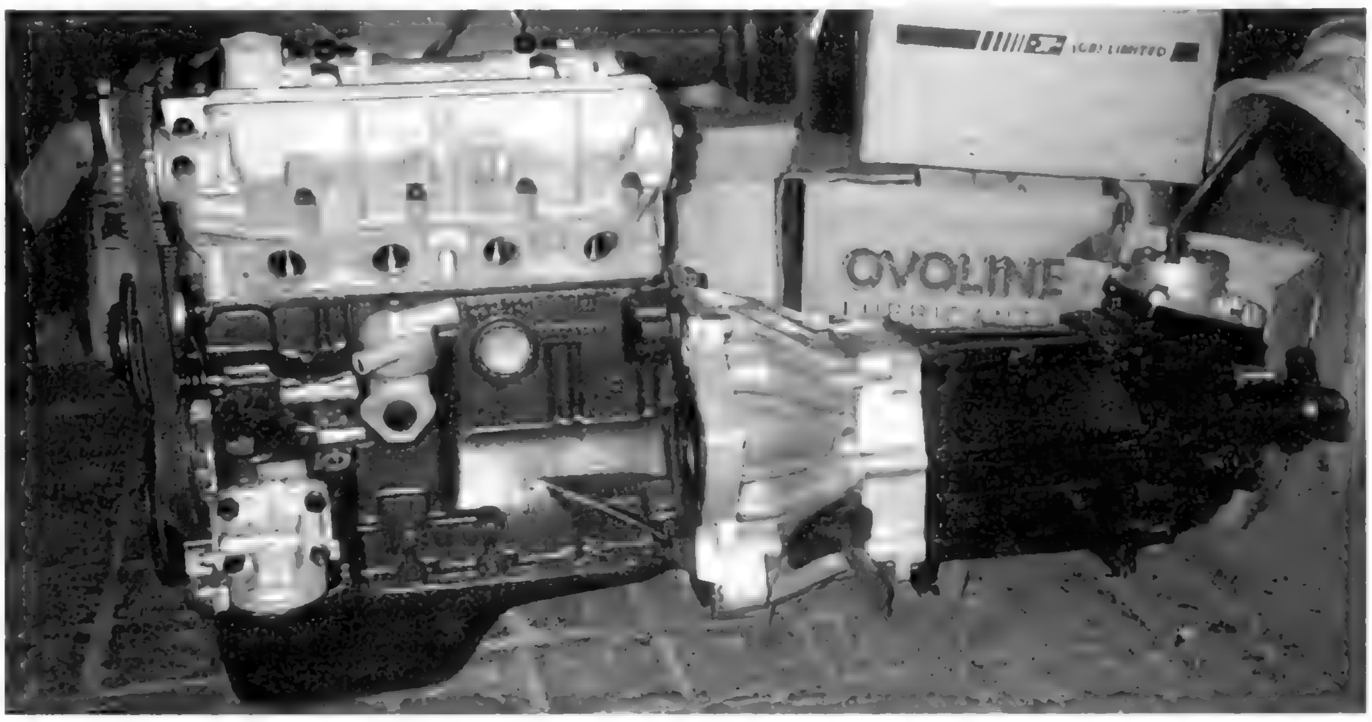
1st	1.521:1
2nd	1.323:1
3rd	1.150:1
4th	1.000:1
5th	0.870:1



17/11: Author's 124 CSA 1800 motor in 1985. Late-pattern 131 2l bellhousing and box fitted. Engine has Titan Series 1 three-stage dry sump. Of course, when engine was trial-fitted, sump fouled crossmember!



17/12: Clever 5" extension to remote gearshift on 131 Sport box, developed and made by Martin Gallacher for this Westfield 2l. Martin used a Cortina IV/V propshaft (perfect length) with Fiat spline pressed and welded at gearbox end.



17/13: Gareth Jones' 1600 Fiat shows what can be achieved, DIY, with modest facilities; 2l is mated to 2l 131 (4-door type) gearbox. This version does not have Abarth-designed remote gearshift.

In practice, 1st and 2nd tend to be lowered to improve the startline performance, and a slightly increased power drop will be tolerated between 2nd and 3rd. A rally gearset might use an r value of nearer 1.25.

Differential ratios

TYPICAL AXLE RATIOS USED IN TCs AND CONVERSION CARS		
124 ABARTH RALLY 1800	10/43	(4.3:1)
124 BC 1608	10/43	(4.3:1)
132 1800 GLS	10/41	(4.1:1)
132 2*l* (MANUAL) (AUTO)	11/41 12/41	(3.73:1) (3.42:1)
131 1600 2000 (RACING) 2000 (4 -DOOR)	12/47 12/47 10/35	(3.9:1) (3.9:1) (3.5:1)
130TC	15/51	(3.4:1)
105TC	17/61	(3.588:1)
LANCIA BETA 1600 2*l* MONTE CARLO VOLUMEX	15/67 14/52 19/62	(4.67:1) (3.78:1) (3.71:1) (3.26:1)
FORD SIERRA Type N (IRS)	VARIOUS RATIOS AVAILABLE, *eg* 3.77:1, 3.62:1, 3.38:1	
FORD ESCORT (75–80)	4.12:1, 3.89:1, 3.54:1 (TIMKEN) 4.44:1, 4.11:1, 3.89:1 (SALISBURY)	

With the exception of the 124 Abarth (and of course FWD models) and Ford Sierra, all these axles are 'live'. It is worth noting that the final-drives in the Beta boxes are interchangeable.

GCT usually recommend a final-drive ratio of around 3.3:1 for high-torque engines requiring a high top speed, and 4.2:1 for engines requiring high acceleration but modest speed.

Relationship between tyre/wheel size, differential ratio and engine speed:
Differential ratio: Rd
Gear ratio in top: Rg
Engine speed at max power: n (rpm)

1 – Calculate tyre thickness T (in):
Tyre type normally quoted as width × aspect ratio, *eg* 205 × 50, *ie* 205mm wide, 50% aspect ratio. This gives a value for T as 205 × 50% = 102.5mm (divide by 25.4 to convert to inches). Therefore, T = ...(in)

2 – Calculate overall tyre and wheel diameter D (in):
D = wheel dia (in) + 2T. Therefore, D = ...(in)

3 – Calculate axle speed at max power N (rpm):
N = n × Rd × Rg. Therefore, N = ...(rpm)

4 – Calculate tyre and wheel circumference C (in):
C = π × D = 3.142 × D. Therefore, C = ...(in)

5 – Calculate road speed V (mph):
V = N × C × 60 ÷ 63360. Therefore, V = ...(mph)

A typical example might be:
Vehicle: 124 Abarth Spider
Engine: n at max power = 6200rpm
R_d = 10/43
Rg = 0.881
Tyres: 185 × 70
Wheels: 13" dia

1 – T = 185 × 70%
= 129.5mm
= 129.5/25.4in
= 5.1in

2 – D = 13 + (2 × 5.1)
= 23.2in

3 – N = 6200 × 10/43 × 0.881
= 1270.3rpm

4 – C = π × D
= 3.142 × 23.2
= 72.89in

5 – Hence road speed at 6200rpm

$$= \frac{1270.3 \times 72.89 \times 60}{63360}$$

= 87.7mph

Use of a 10/39 diff would raise the speed at 6200rpm to 96.7mph as it would raise the N value to 1400.5.

Limited-slip differentials

Limited-slip units greatly improve traction during cornering and should always be considered for a serious competition car. The difficulty in procuring these for any but the latest Fiat/Lancia TC models (not, unfortunately, including the entire Beta range, 131 or 132) means that the TC owner may inevitably have to construct a hybrid vehicle using, perhaps, one of the Ford axles listed since LSDs are still readily available for these. Regulations should be carefully examined to ensure that the final layout is 'race legal'.

TRAN-X
The name Tran-X, of Redditch, is synonymous with expertise in supreme competition gearsets. For Ford-Fiat conversions the two following will be of interest:

5sp Sierra Type N or 9 (Rally)

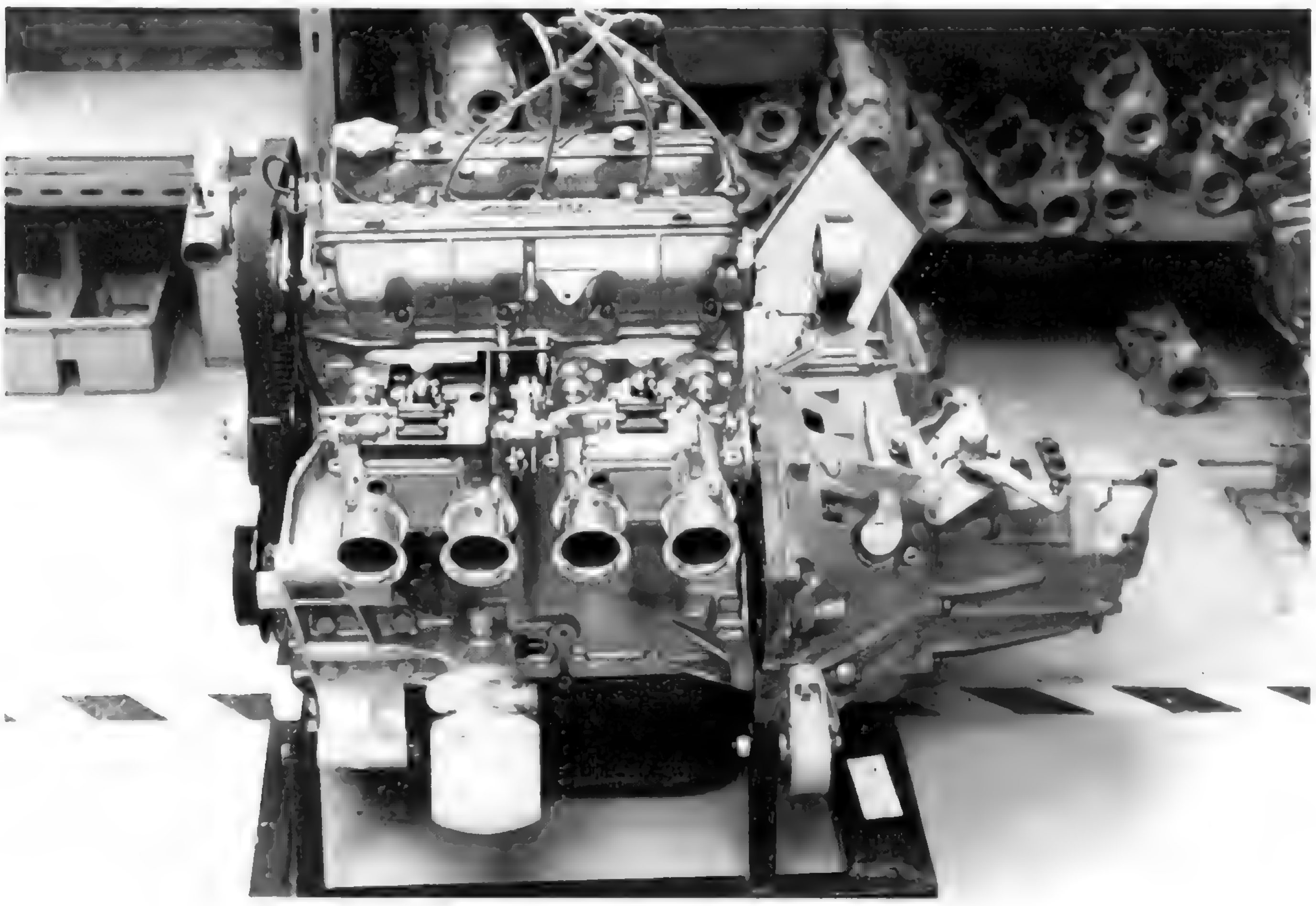
		From	
1st	2.48:1	Transit	11/85
2nd	1.69:1	Sierra	8/82
3rd	1.27:1	Capri	11/82
4th	1.00:1		
5th	0.87:1		

4sp RS 2000 (NHRA)

1st	2.48:1
2nd	1.69:1
3rd	1.61:1
4th	1.00:1

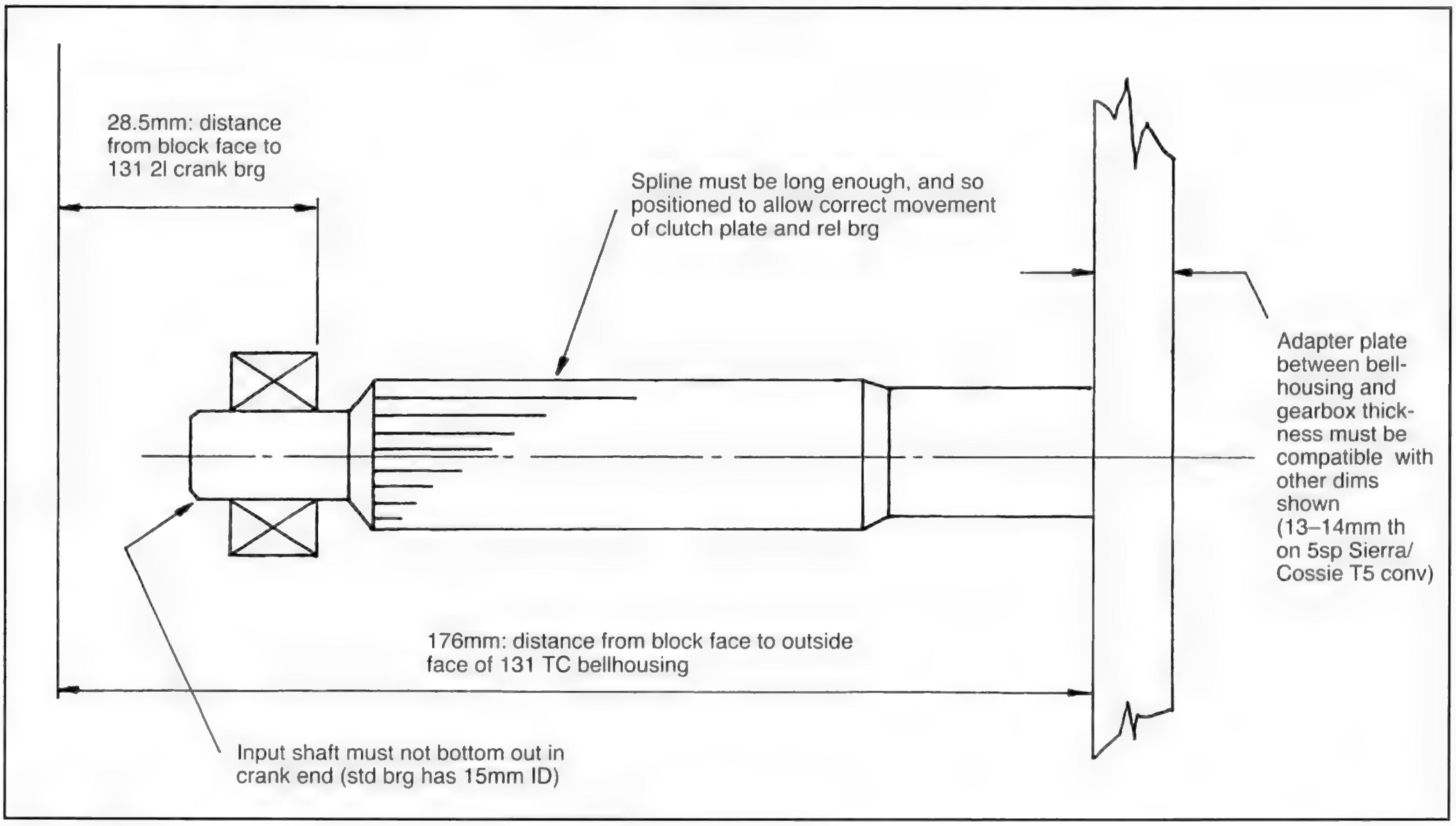
These sets are straight-cut and retain the standard synchro unit; a needle-roller input shaft conversion is advisable. Tran-X also specialize in LS units and have the manufacturing rights for the famous Ford Powrlok unit. Among their projects to date is a plate diff for the 16v Tipo.

Additionally, they have produced synchro gearsets (with a quick-change casing) for the Gp A Integrale. Tran-X also have the in-house facilities to manufacture driveshafts to order.



17/14: GC St II Lancia Monte Carlo engine and gearbox. Compact nature of complete power-pack shows why so many manufacturers adopted unitary layout. Note straight-shot inlet manifold with top-mounted distributor – grafted from 1608.

17/15: Key data for adaptor plate conversion using 131 TC gearbox.



CHAPTER 18

OWNERS' CARS

One of the great strengths of the Fiat/Lancia TC is its sheer versatility as a powerplant. Part of its appeal is the prodigious power output that can be realized for a fairly modest investment, and part is due to the compact nature of the engines and the relative simplicity of their layout.

Despite being fundamentally derived from a design over 20 years old, these engines are still much in demand as clubman competition power packs, and when used in original vehicle applications, still represent exceptional performance when even mildly tuned.

18/1: Centre and right, two 131 Sports owned by Oxfordshire-based Simon Reeves and, left, car belonging to his friend Gareth Kerry. A sort of micro 'owners club' typical of Fiat TC owners around the country.

18/2: Meet a tuned 131 Sport on the road and this is the view you are most likely to see! Phenomenal torque (123lbf ft) makes car extremely quick, even by '90s standards.

18/3: Tidy interior of Simon Reeves' 131 Sport. Engine has 9.6:1 CR, lightened flywheel, gas-flowed head (42/36 valves), 45s (36 choke), circa 155bhp @ 6300rpm, 142lbf ft torque @ 3600rpm. Flat torque curve gives terrific acceleration through gears. Air filter is one-piece ITG. Backplate keeps carbs in balance. Just visible is single coolant hose from head to radiator. 'In-head' thermostat has been fitted; external production thermostats run too hot for modified motors.

18/4: Simon Reeve and Gareth Kerry write: "We have listed a few things that may be of interest to Fiat 131 owners. All of the following have been successfully carried out by Gareth and me and do work.

1 When changing axles it is worth noting that those from the series 2 131 (post-1982) are 17mm wider each side making the total track 34mm wider. This could cause the tyres to rub the inner arches if wider wheels are fitted, especially if the suspension has been lowered; 6in rims are about the maximum width without rear arch modification. It also follows that the halfshafts and diff units are not interchangeable between series 1 and series 2 axles.

2 We have converted 131 Sports to power steering by using Supermirafiori 2000 crossmember, rack, steering shaft and front pulley and adapting the engine mounts. Some 131 Sport blocks allow the later engine mounts to be bolted straight on as they have six studs; if the block is of this type the spare holes must be plugged with M8 bolts to prevent oil leakage. When this conversion is done the wheels will have more negative camber as the mounting points for the track control arms are 15mm more outset each side on the Supermirafiori crossmember, the resulting track being 30mm wider.

3 Fiat 131 brakes can be uprated by using discs and calipers from Regata 100 or Lancia Prisma which bolt straight on. These use the more reliable sliding pins rather than wedges. Further to this, Uno Turbo vented discs and calipers can be used.

4 When shimming a Twin Cam in the car, it is useful to pump the oil from the cam carriers; we use a fuel pump from an injected Volvo with crock clips attached to the battery."

18/5: Ray Carden's 1989 Mazda 323, powered by a home-built Fiat 2l (twin 45s), convinced him that engine had real potential in Autograss (grasstrack) racing. Next car (not shown) was a mid-engined Fiesta driving through an Audi transaxle gearbox.

18/6: By 1992 Carden had opted for a GC engine (No 142) fitted in a Metro. Note front and rear wheel sizes!

18/7: A 2l Lancia layout cleverly grafted to front of Metro. Although engine layout was changed when Cardi Racing adopted a Starlet in late '92, engine remained essentially the same, with different sump, cam boxes and change of crank bush to bearing for RWD gearbox. Note use of Fiat coolant outlet elbow on head. Output was around 160bhp @ 6300rpm with standard 2l cams. Radiator is in rear of car.

18/8: Car No 4! Toyota Starlet with Fiat set-up. After early promising results, cams were swapped (in situ) for Guy Croft St III rally types; car started to win.

18/9: By 1993, Ray Carden and his Cardi Racing team had graduated to NASA Class 3 Toyota Starlet equipped with full-spec St II 2l, GC engine No 142/3. Ray, his son Jason, and engineer Peter Smith took nine months to prepare car; result certainly justified effort, as these pictures show. One of best-prepared grasstrack cars around.

18/10: Half armoured car, half greyhound! Cars are built with safety in mind as racing on a grass oval is fast, furious and unpredictable. Class 3 is over 1351cc, two valves per cylinder, normally aspirated engine and rear-wheel drive.

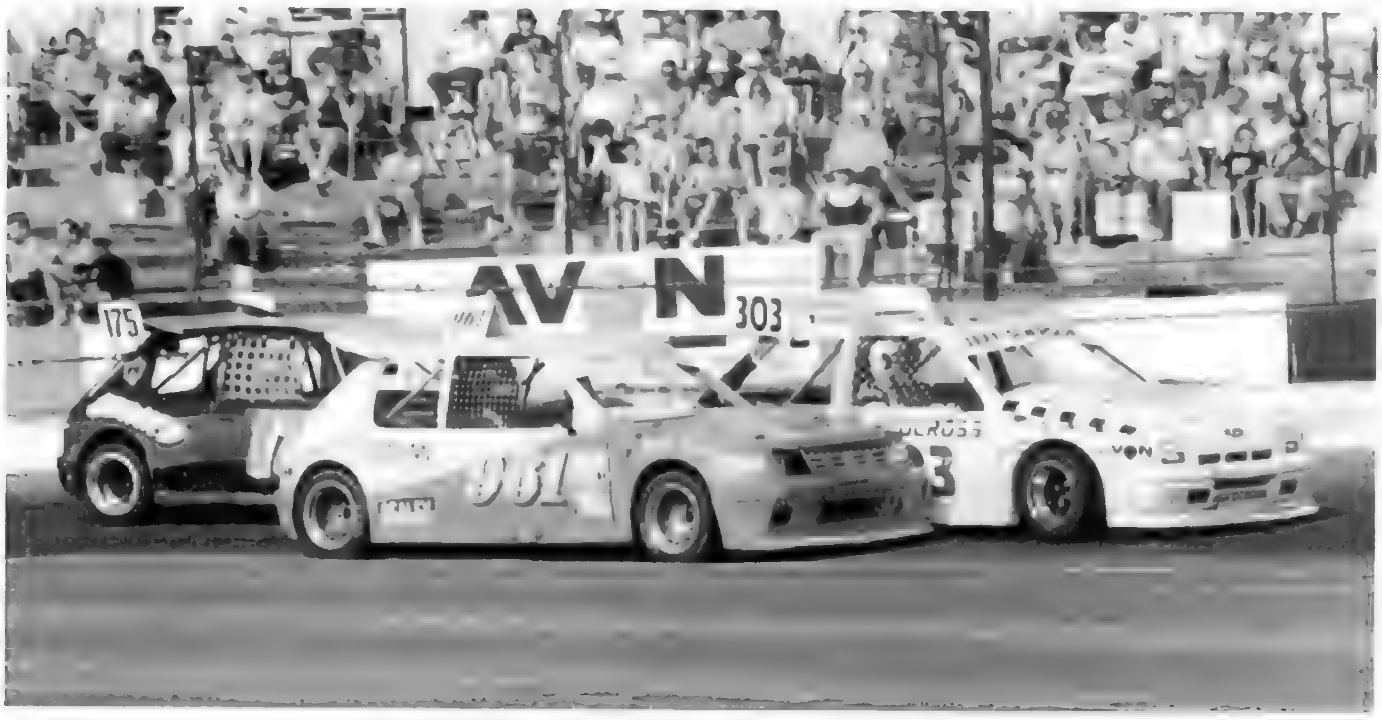
18/11: Engine has 10.2:1 CR, St III rally cams, 45 DHLAs (38 chokes), 42½/37 valves, and is fully ported and blueprinted. Special sump with rear well, boxed/baffled, clears Toyota cross-member. Air filter (not shown) is ITG. Engine develops circa 176bhp @ 7000rpm; mated to 5-speed Sierra box via GC adaptor and AP Racing RS 2000 competition clutch. Rear axle is 3.9:1 Capri Atlas with LSD. Braking is by 8.6" Ford discs all round; 200lb front, 130lb rear springs with 4-link rear end.

18/12: Class 3 Champion Jason Carden with car at Mid-Kent Autograss Club track at Ivychurch, Kent. Jason won Class 3 Championship at Mid-Kent in 1993, also Saloon Car Shield (for classes 3, 4, 5 and 6!). At Hot Rod Enterprises track, Headcorn, Kent, also took 3rd in Class 3 and won SRF 6 trophy.

18/13: Competition! Lightweight special nips inside the Cardi Racing Starlet coming out of a turn. Illustrates how hard team worked to achieve their outstanding '93 results. Car races in only 2nd or 3rd gear so broad spread of torque is vital.

18/14: Left to right, Ray, Peter, Jason and Ray's fiancée Michelle with car and their '93 trophies. Driving at Mid-Kent track, Shelley won '93 Class 3 Ladies' Club Championship and came 4th overall. Grasstrack is deliberately structured to give as many races (and chances) to all comers and is consequently increasingly popular.

18/15: Tom Casey with his National Hot Rod at 1994 World Hot Rod Championship. Engine (see Case History No 1) was home-built full-race 2l Fiat driving through 5-speed straight-cut Ford gearbox. Some idea of quality of competition can be gained from this view of cars in paddock. Note huge split-rim Compomotive wheels and slicks: NHRA racing requires the most competitive set-ups money can buy. (Photo Chris Berry)



18/16: Ipswich: Tom Casey trying to hold the inside line against Jeff Simpson's quicker spaceframe car. (Photo Chris Berry)

18/17: Tom Casey is a study in concentration waiting for start flag. Indoor racing (pre-Fiat) at Earls Court during '93. (Photo Chris Berry)

18/18: Delighted Tom Casey, '93 Irish Hot Rod Champion, receives '94 World Championships 4th Trophy from former World Champion George Polley, designer of famous Polley Rods – Starlet-based rods which revolutionized the sport in '80s. (Photo Chris Berry)

18/19: Tom Casey proudly displays trophy outside GCT following '94 World Championship at Ipswich. No engine in car! This was removed after race for testing (see Case History No 3). Tom drove with consistency and skill, avoiding numerous crashes and exclusions during 70-lap race to achieve well-deserved placing. A hugely popular driver, Tom often visits UK from native County Waterford. Performance of his home-built engine (GC components) was exceptionally reliable, but lack of 'punch' out of corners led to engine being redesigned at GCT in late '94 (see Case History No 2). Car is Peugeot 205 GTI with spaceframe welded to existing floorpan. Front suspension is MacPherson strut, rear is live axle, mandatory in NHRA. Panels are Kevlar or glassfibre. Although a bit on heavy side, a well set-up car which handled very well despite weight and chassis flex compared with fully spaceframed cars.

18/20: Former British Champion National Hot Rod driver Mick Phillips, of Birmingham, dabbled briefly with his own 2l big-valve (46/40) Fiat during '94. Pictured at Ipswich '94 World Championship, Mick's car suffered from plethora of petty problems; breather trouble, incorrect cam timing, overheating, throttle linkage falling off, etc, and although potentially one of quickest cars on circuit, never realized its full potential. Owner ditched home-built engine in favour of Vauxhall 16v at end of year before any serious development could be carried out. Note large cold air duct above carbs; unless air is also ducted internally direct to carbs, high-pressure air under vehicle may cause air to flow out of duct rather than correct way, or stall in duct and go nowhere. High underbonnet air temperature robs bhp. Chassis is purpose-built SHP 'Peugeot 205 GTI' spaceframe, with glassfibre/Kevlar body.

18/21: Toyota Starlet converted to National Hot Rod spec and owned by Paul Thomas from Woodseaves, Staffs. A former Polley Rod, this car was originally raced by George Polley himself, though not with a Fiat engine! Still a quick car, but not competitive against the latest speceframe designs. It is pictured at Hednesford Raceway during summer '94, by which time it had been fitted with updated engine No 212 (see Case History No 7).

18/22: Hednesford Raceway 1993: Paul Thomas dicing with Joe Tandy just before a turn. Front-end damage was caused earlier in race.

18/23: The first GC engine raced by Paul Thomas. St IV sprint-race cams, 10.5:1 CR, 43.5/36 valves, sidedraught 45 DHLA carbs (40 chokes), diecast pistons, steel flywheel and race sump. Output in region of 178bhp/146lbf ft. This engine was wrecked when a stone caught under cam belt in early '94. Paul had 'spaceframed' car and slick tyres showered engine bay with stones during cornering; simple shields around wheel would have prevented this. GC Eng No 189.

18/24: Paul Thomas racing at Incarace oval circuit at Hednesford, Staffs, during late '93. National Hot Rod is without doubt the most competitive form of racing in UK. Extremely powerful engines and superlight cars with exceptional handling always give a highly packed, thrilling race. Here 'Jaffa' Thomas is closely tailed by Ricky Hunn, who became World Champion in '94.

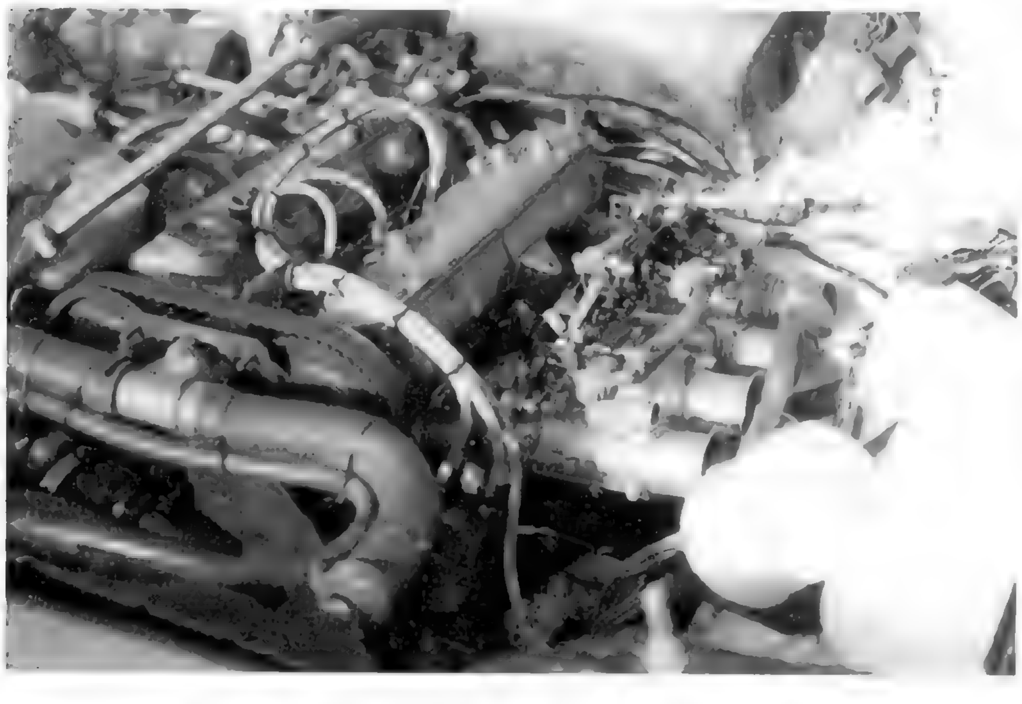
18/25: Bumper to bumper, three-abreast racing into turn usually leads to demise of one car or another. In this case it was Paul Thomas! National Hot Rod is non-contact racing, but when quicker cars are held up, tensions rise, tempers flare and contacts are guaranteed, to delight (or fury) of supporters in stands.

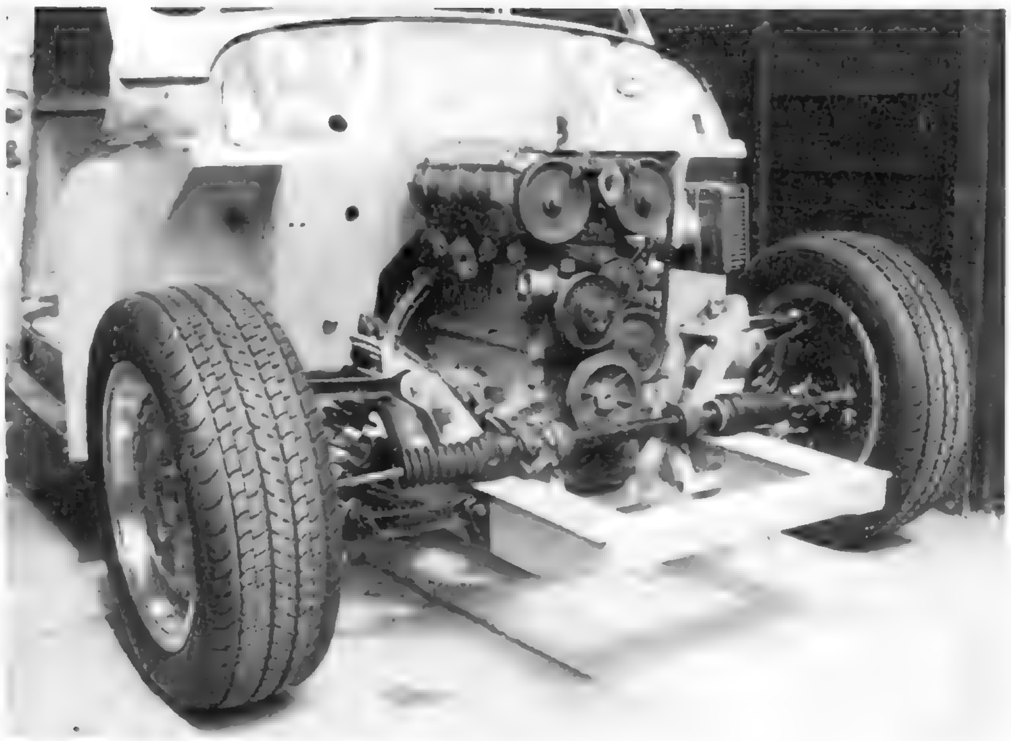
18/26: Jonathan Douglas, director of ITG (air filter specialists) with his 4/4 Morgan at Silverstone in 1992. At this time, engine was a home-built 2l. ITG are a leading authority on intake and filtration systems, producing superior air filters for Grand Prix cars or clubman racers with equal devotion. (Photo Fred Scatley)

18/27: Colin Rowe's outrageous pink 131 2l. This beautifully prepared rally car was restored and prepared by Colin 'from the ground up' and was featured in Cars & Car Conversions magazine in early '94.

18/28: Colin has now converted his 8v 131 2l engine to accept late (Thema/Integrale) 16v head to boost power to 200bhp-plus range, with GC cams, forged pistons and 45s.

18/30: One step further with Paul Empson's Fiat-Morris Minor. Radiator and oil cooler fitted. Note GC offset sidedraught manifold used with block-mounted distributor.



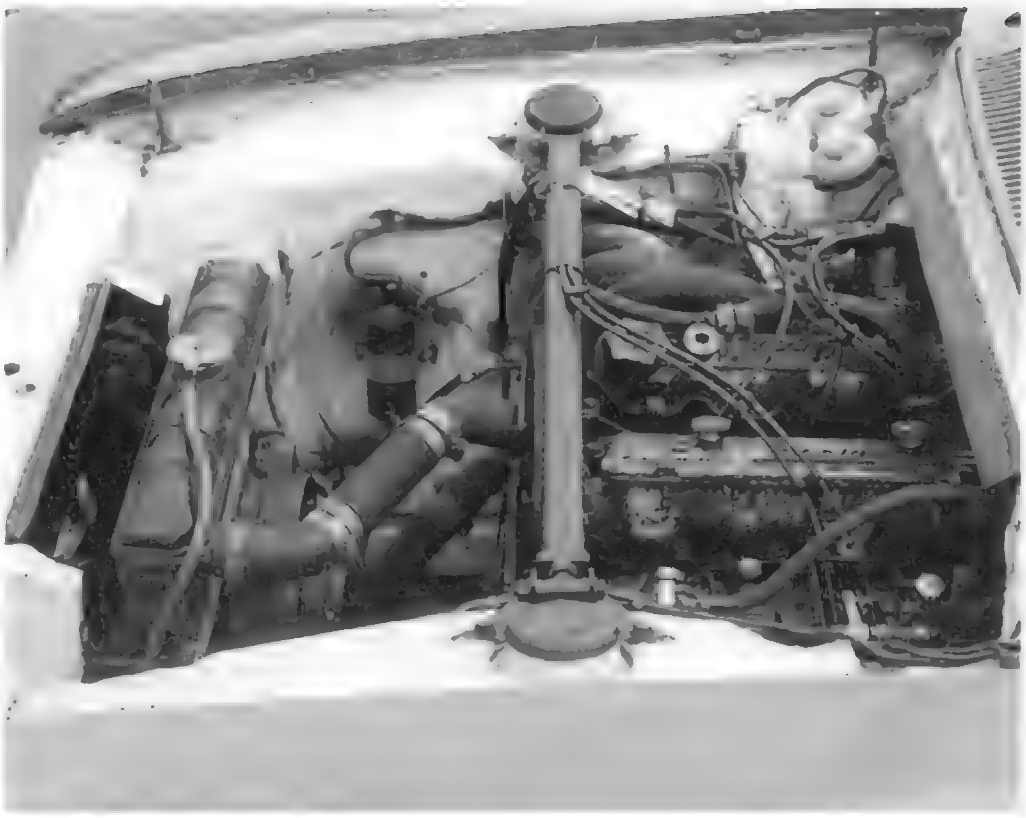


18/29: Paul Empson's Fiat-Morris Minor conversion mocked up with 1600 unit. Judging by quality and extent of preparation so far, this car should be stunning when finished with 2l engine.

18/31: US-born Julian Sudano, restaurateur from Glasgow, poses with his beautiful 1800 Spider. Originally a strangled low-compression US engine, this unit was converted to Euro pistons (9.8:1 CR) and cams, plus downdraught 40 IDF carb. A desirable example of a now rare and appreciating sportscar.

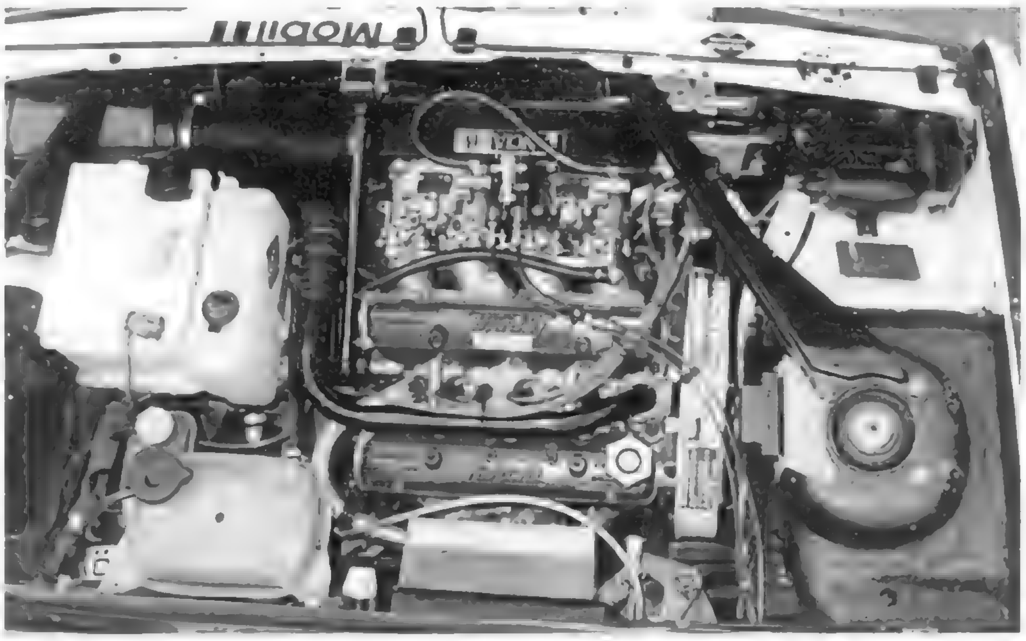
18/32, 18/33: Alan Cooper's Vauxhall Chevette – 2l Fiat conversion. This vehicle is tailormade for Fiat TC conversion. Note remote filter mounted on right-hand inner wing. Filter used in this inverted layout must be fitted with non-return flap valve to prevent oil draining out of circuit when engine is switched off.

18/34: Gorgeous, boxy Mk 1 Fiat-Escort made brief appearance at GCT during 1988. Engine mounted behind crossmember allowed retention of standard Fiat sump. Although motor intruded heavily into tunnel, handling was superb with weight concentrated towards centre of car.



18/35: Escort radiator, 1608 engine (top-mounted distributor), side-draught 40s with cleverly executed 4-1 exhaust made for a very rapid car, ideal for clubman rally; 1608 had 80x80mm bore/stroke, steel crank. Lovely free-revving power unit.

18/36: Tim Walker's beautifully maintained 2l Beta Spider features GC ported/blueprinted head, St II cams, 40 sidedraughts. Power in region of 155bhp at flywheel.



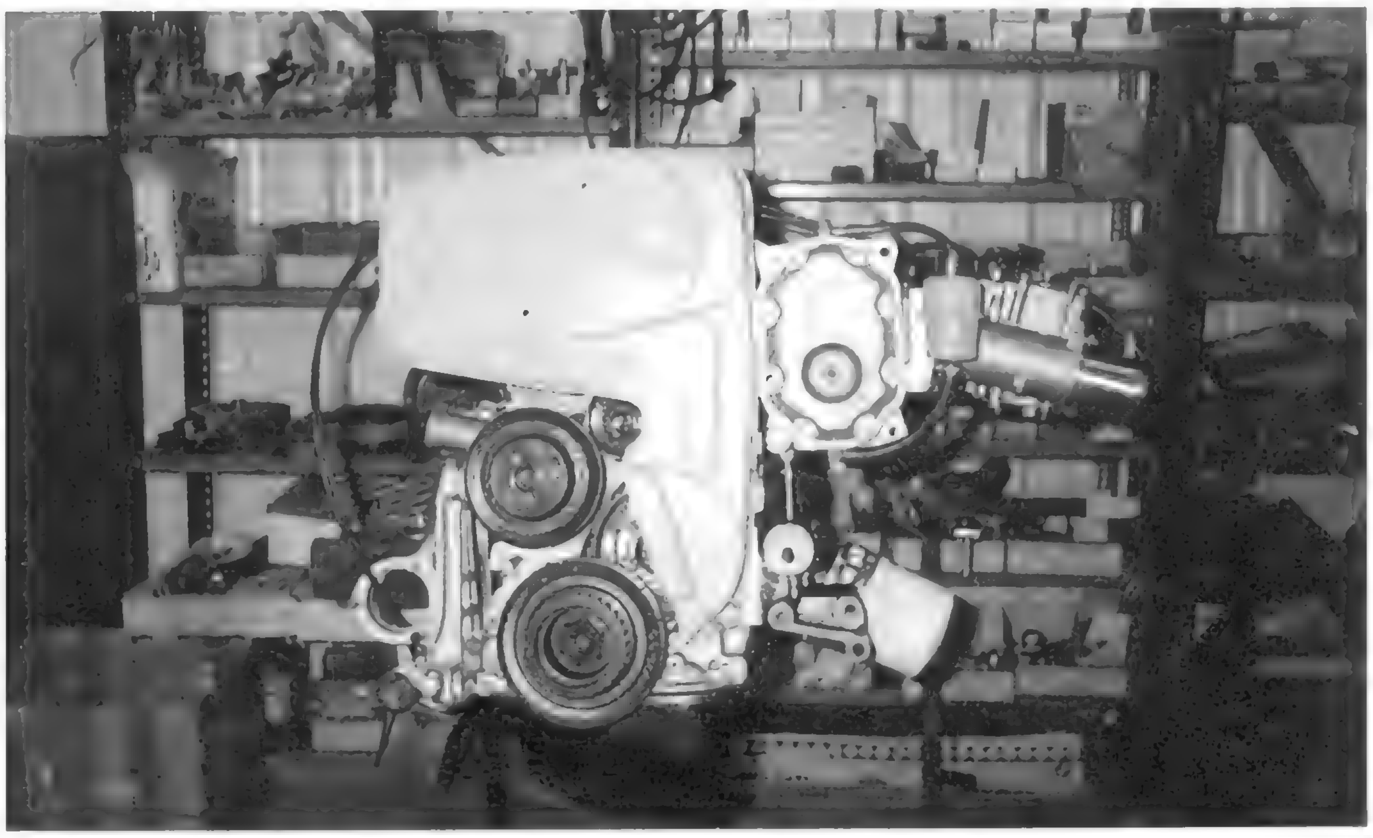
18/37: Well turned out 2l Lancia Beta engine belonging to Mario Grech-Xerri. Full spec included 10:1 CR, sidedraught 45s (38 chokes), St III forest cams, ported/blueprinted head, 42½/37 race valves, lightened and balanced crankshaft/flywheel assembly, Gp N clutch. Note cold-air box to carbs made from Peugeot item. Power conservatively estimated from rolling-road dyno test at 180bhp at flywheel.

18/38: Well worn 2l Lancia belonging to John Day photographed during 1990. Dry-sumped unit was essentially same as Tom Casey's (see Case History No 3) but more competitive in circuit racing: engine took Day to countless class wins/records over three-year period in Italian Intermarque Challenge. Reliability of this GC-built engine was demonstrated by fact that it raced in '92, '93 and '94 without any maintenance at all (and was still producing competitive 188bhp by mid-'94)! Cheap convoluted hose on coolant circuit is 'no-no' on race engine – very prone to splitting. Amazingly, fine-threaded oil pressure sensors screwed directly into top of remote filter head never blew out! Note substantial sway brace between suspension pillars, essential to retain chassis torsional stiffness after removal of front end of shell. Deflector on carbs helps to prevent warm air from radiator feeding carbs – bhp loss.

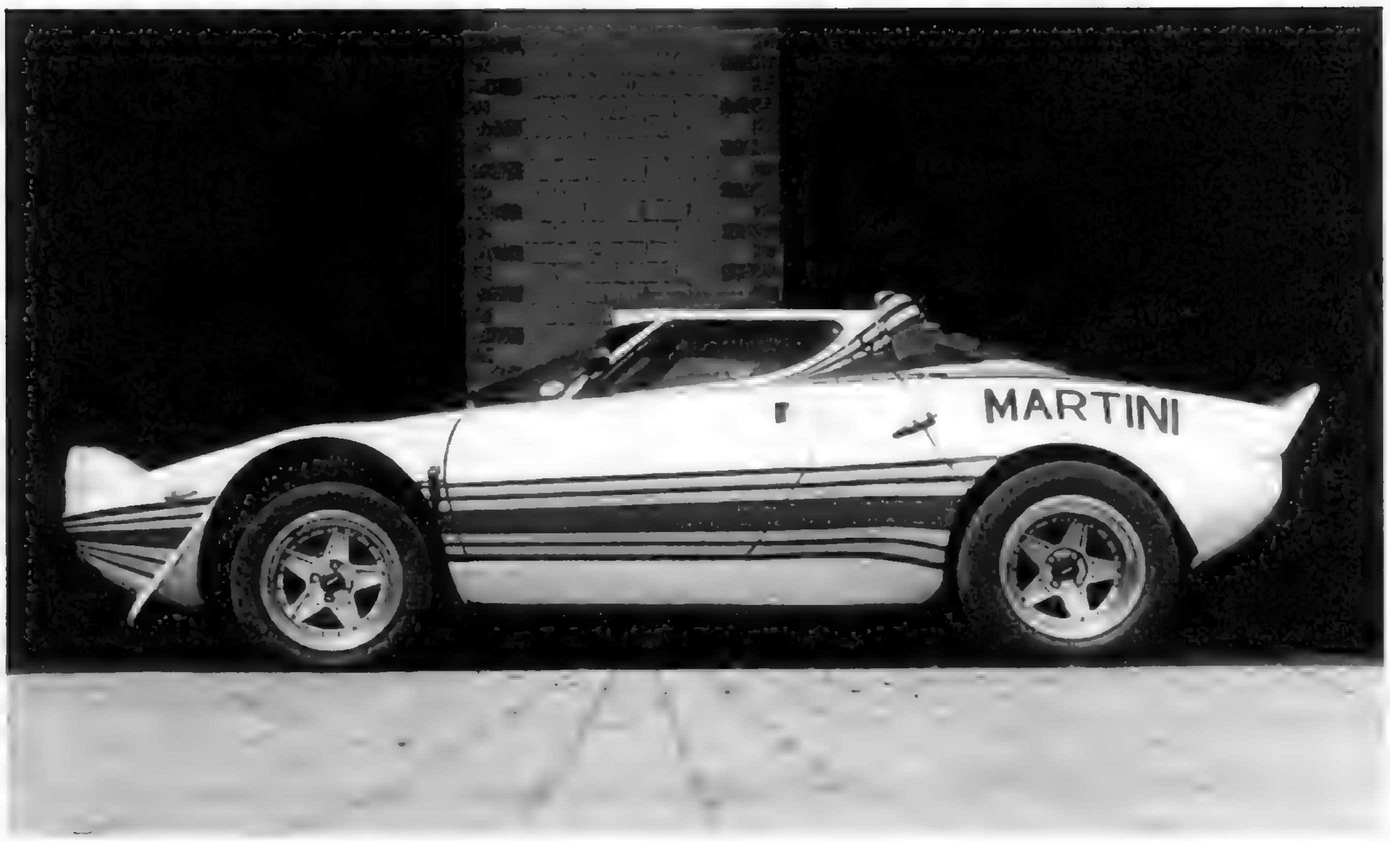
18/39: Alfa Beta! John Day in characteristically aggressive pose leading Italian Intermarque Challenge race. John is one of quickest drivers of front-wheel-drive machinery in UK. Long-legged engine and low, swoopy car add up to dangerous combination!

18/40: John Day sweeping to victory in another Italian Challenge race. Note redesigned/extended front spoiler. Day was outright Class B winner in '91 with two overall and eight class wins and six lap records! Engine had 10.5:1 CR (forged pistons), 46/40 valves, dry sump, steel flywheel, 7¼" clutch, St IV race cams; 48 carbs (42 choke); developed 198bhp @ 7500rpm.

18/41: Fiat 1608 trike owned by Richard Humby features downdraught 40s, 124 5-speed gearbox and Reliant back axle. Gearchange effected via long chromed lever attached directly to gearbox! This unique machine has been driven (ridden?) all over Europe by Humby, who built it. Handling (and engine note) superb!



18/42: Build-up 2l Volumex unit with side-draught 45 DCOE for Jerry Smith at GCT. Unit awaits direct-drive belt conversion. Sump is GC race-spec; bearing life with standard sump at 180lbf ft-plus torque could be measured in minutes in even fast road use!

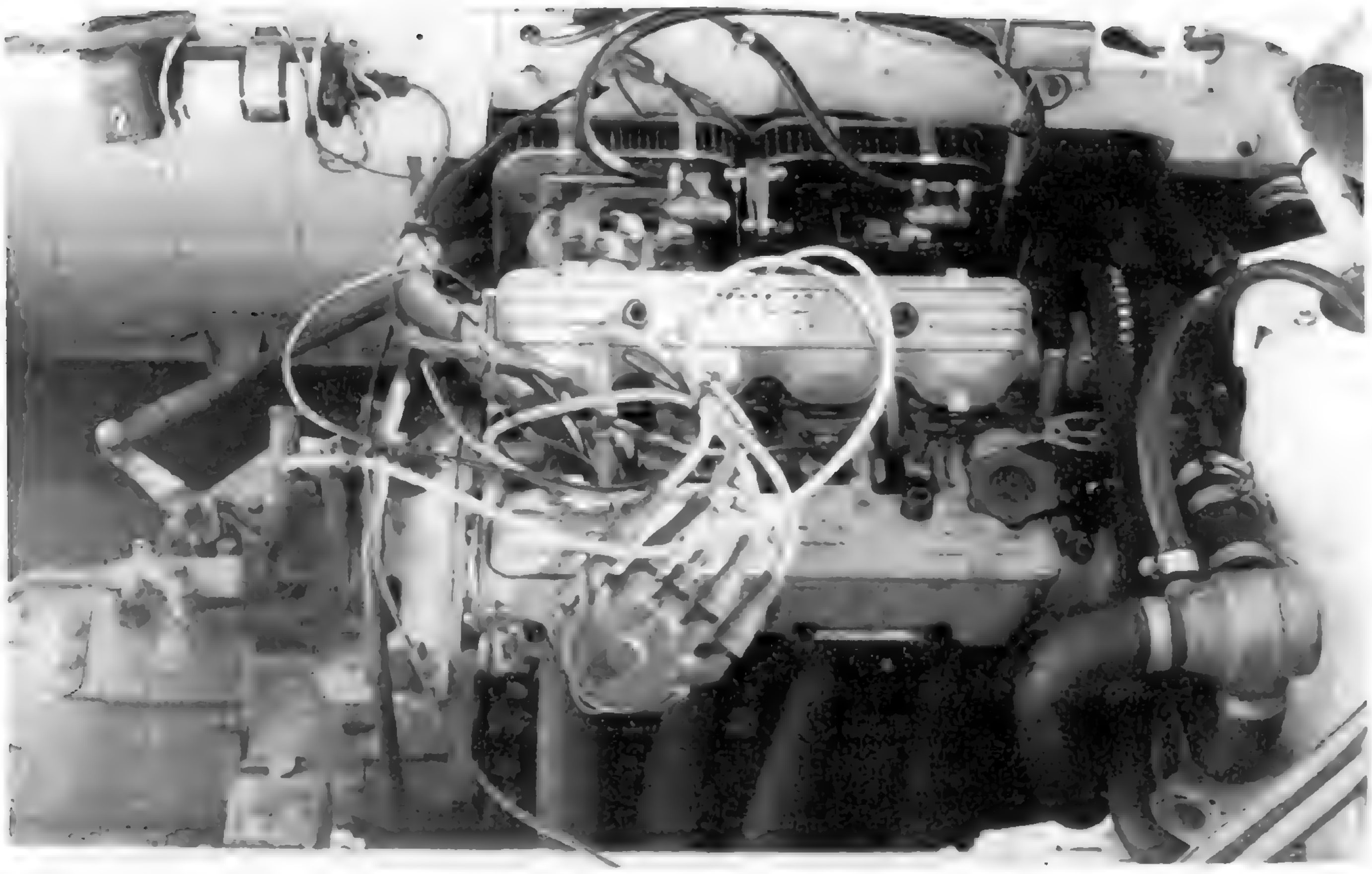


18/43: Supercharged Stratos replica of Pete Luxford features Guy Croft engine type as in picture 18/42 with around 215bhp at flywheel. Acceleration produces same physical sensation as driving flat-out over humpback bridge!

18/44: Gp 4 front end. The car the author would most like to own!

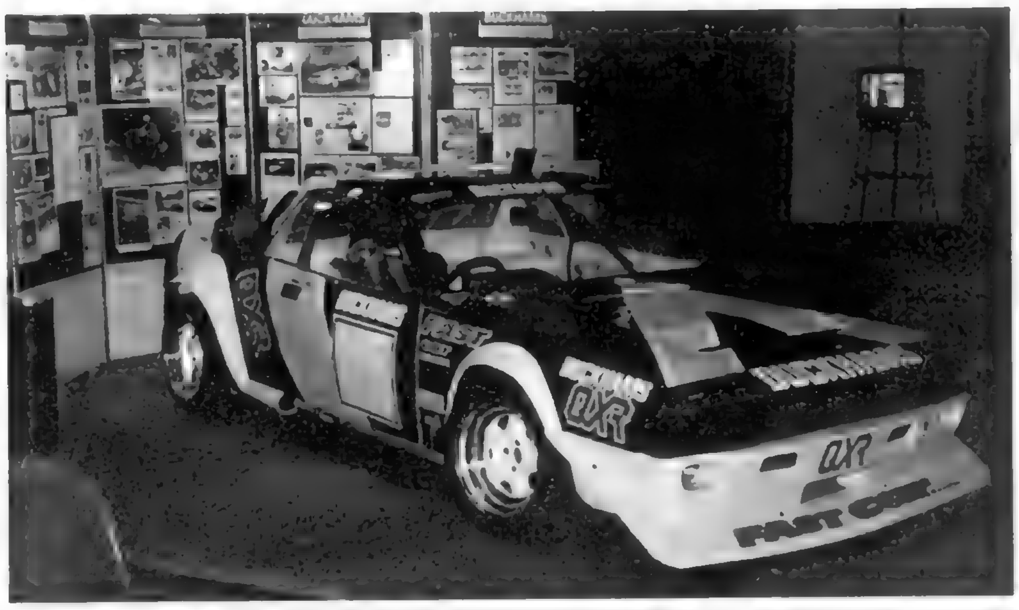
18/45: Luxford's Stratos replica (Gerry Hawkridge design) photographed opposite former Chatham Naval Dockyard on visit to GCT in 1993. Built by Peter, car is finished to highest standards. Gp 4 kit from Hawkridge creates stunning visual impact. [Without doubt one of the most sensational kitcars in the world.]

18/46: Key project at GCT during '88 was collaborative project with Duckhams Oil and Fast Car magazine on this race Monte Carlo built for Italian Intermarque Challenge.



18/47: Engine was GC No 36, featuring 46/40 valves, 48 DHLA carbs (42 chokes), 48/67, 65/46 11.3 cams, dual interference springs. Engine used cast pistons for first six races! 4-1 manifold by TTCM worked superbly. Note use of 124 BC distributor and standard ¾" cam belt; Micro Dynamics ignition. Filters are K&N with very short ram pipes. External thermostat was ditched on later engines due to overheating problems with high power output. Subsequent dyno tests with similar engines indicate power output with final spec motor (forged pistons) must have been in region of 198bhp @ 7500rpm, 154lbf ft torque @ 5500rpm. Car was sensationally quick; novice driver finished 2nd (behind Champion John Day) in first race! Gearbox was standard Monte Carlo via Tilton 7¼" clutch. Alquati straight-cut gearset was tried but gave endless problems.

18/48: GC-powered Duckhams QXR/Fast Car Monte Carlo easily beat a TD Motorsport-prepared Ferrari 308 on a round of Italian Intermarque Challenge at Castle Combe. (Photo Mary Harvey)

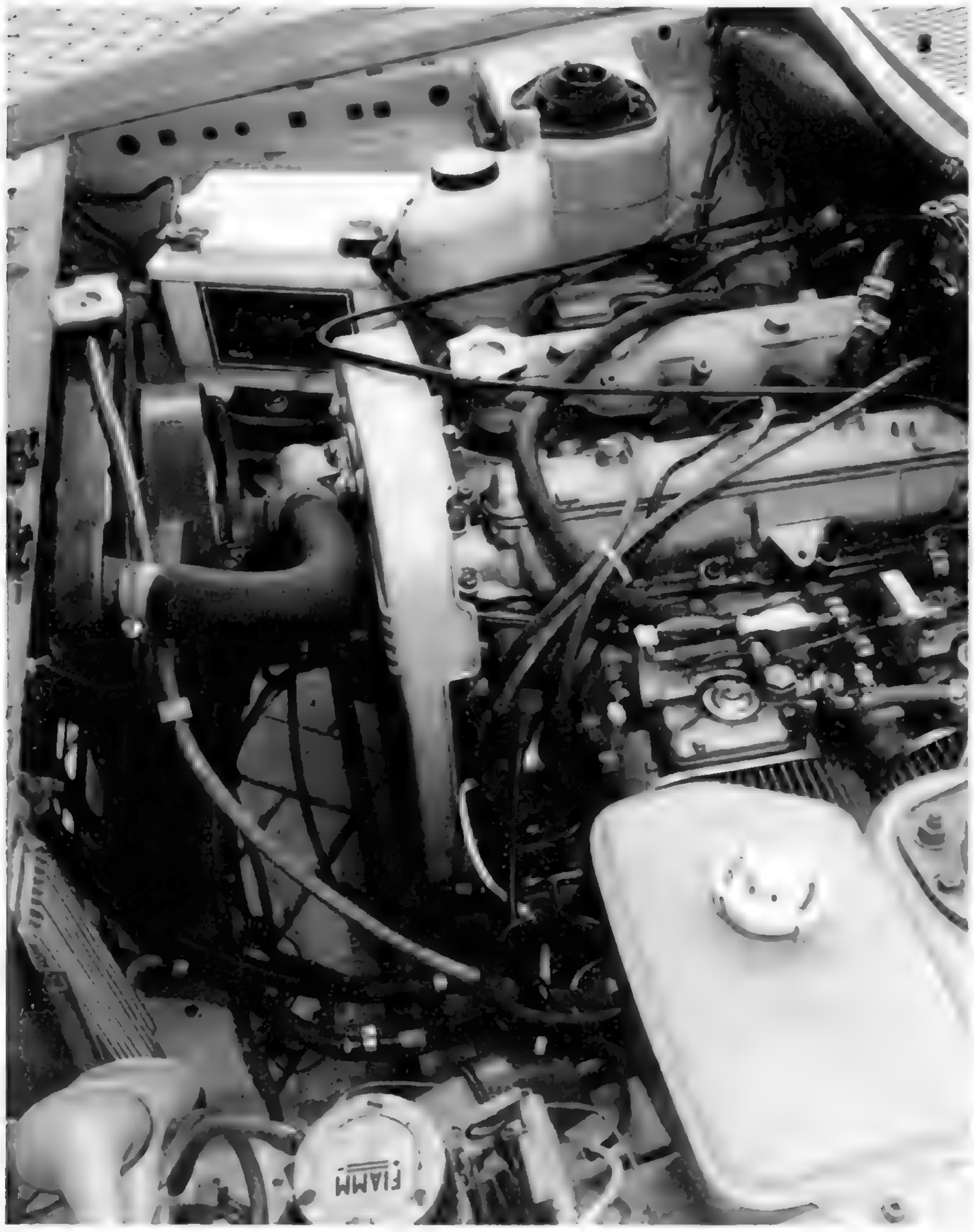


18/49: Beautifully prepared car was featured in series of articles in Fast Car magazine. Project eventually abandoned when owner decided he couldn't afford to continue racing.

18/50: Attractive Transformer Stratos replica prepared by Ian Gear, photographed after return from '91 Classic Marathon. GC race sump was adopted; owner developed clever oil cooler system with ducted electric fan to augment airflow at low speed.

18/51: Anthony Fausset owns this immaculate 131 Sport 2l...

18/52: ...which goes as well as it looks and...



18/53: ...features St II GC 2l engine developing around 176bhp. Carbs are 45 DCOE, 38mm choke. Cylinder head came from early Guy Croft hydroplane engine, featured 42½/37 race valves. Clutch is Gp N Sachs – capable of transmitting up to around 160lbf ft torque. Note modified coolant elbow – engine has 'in head' 74°C thermostat. Standard sump was swapped for big-wing race type. This engine was later transplanted to Lada shown later. Capacious 131 bonnet area makes it popular car for clubman competition.

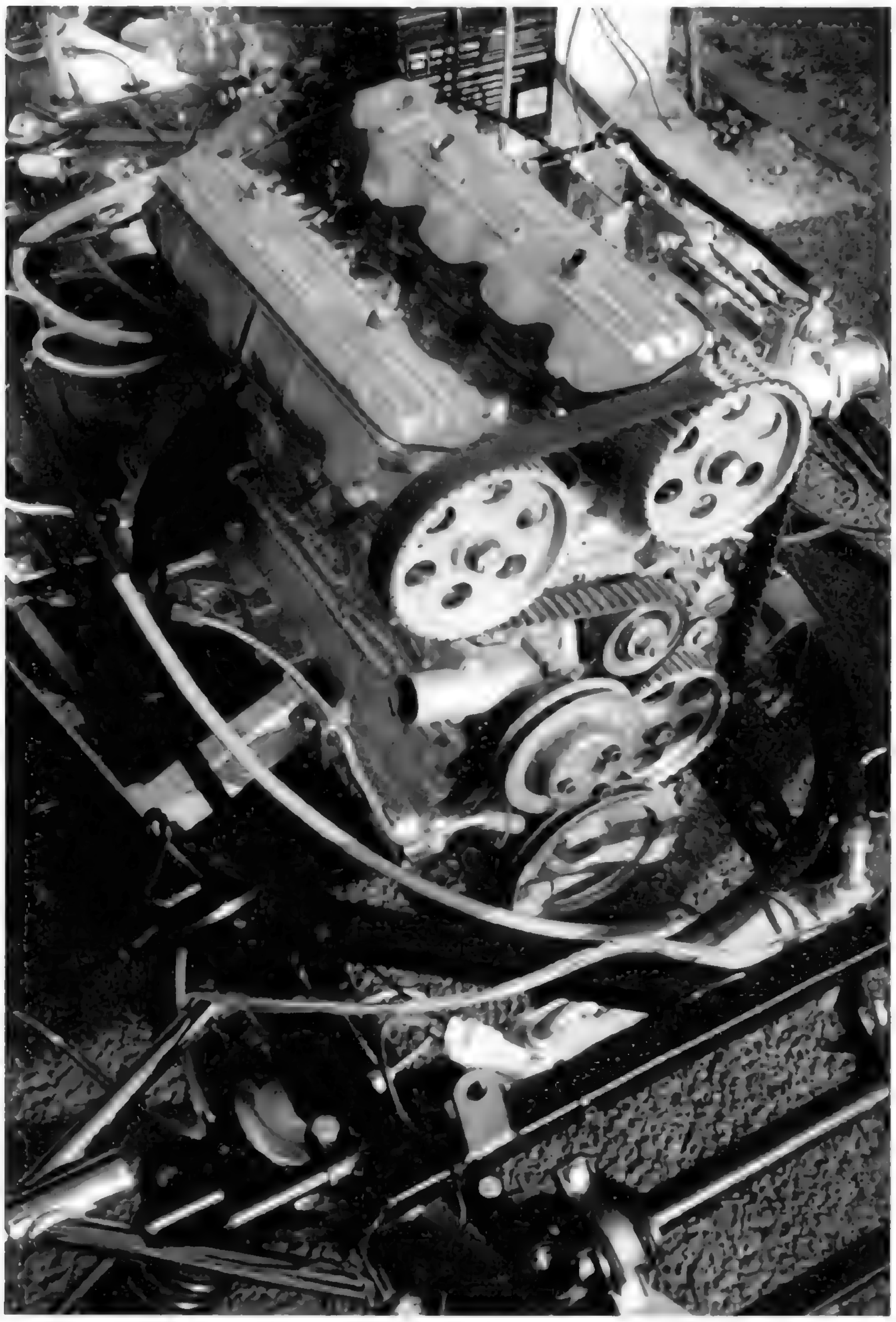
18/54: John Morgan, an engineer based in Thailand, prepared and rallied this 131 in '86, '87. Parts from GCT took engine to turbo spec – 44 inlet valves, sodium-cooled 37 exhausts, St III sprint race cams, twin 48s. Must have produced at least 280bhp on 25lbf/in² boost. Gained regular top placings in Bridgestone – Penzoil events. Clever paint scheme makes four-door look like a two-door!.

18/55: Steve Strain of Wellington, New Zealand, exported this Hawkridge Stratos replica with GC engine No 107. Features fully ported head, 9.6:1 CR, 45 carbs, develops circa 155bhp @ 6300rpm. Looks gorgeous against traditional New Zealand backdrop.

18/56: Mike Rosie of Peterhead, Aberdeen, built this (ex-Litton) Corse with Volumex engine. Engine is completely standard except for addition of 45 DCOE carb!

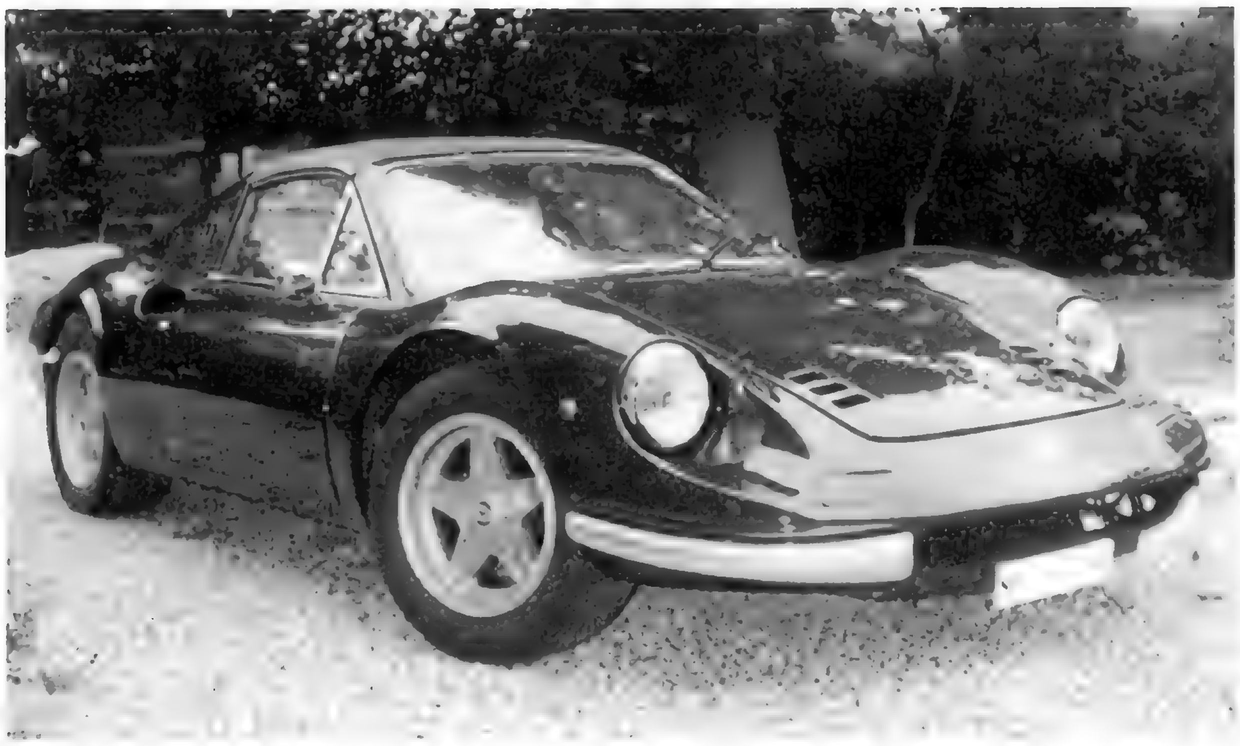
18/57: Mike Rosie and his toys! Volumex Corse has raced against Martin Gallacher's Westfield (below) – no slow car by any standards. In Mike's words: "Corse blew Westfield into the reeds!" Off-the-line acceleration is phenomenal – especially as Volumex, unlike a turbo, pulls right from tickover.

18/58: Outstanding Westfield 2l Fiat of Martin Gallacher. Both men work for Total Oil Marine, at St Fergus, Aberdeen, and along with two other friends, Leslie Swanson and Robert Jardine, share a common love of Fiat/Lancia TCs and meet regularly with their cars.



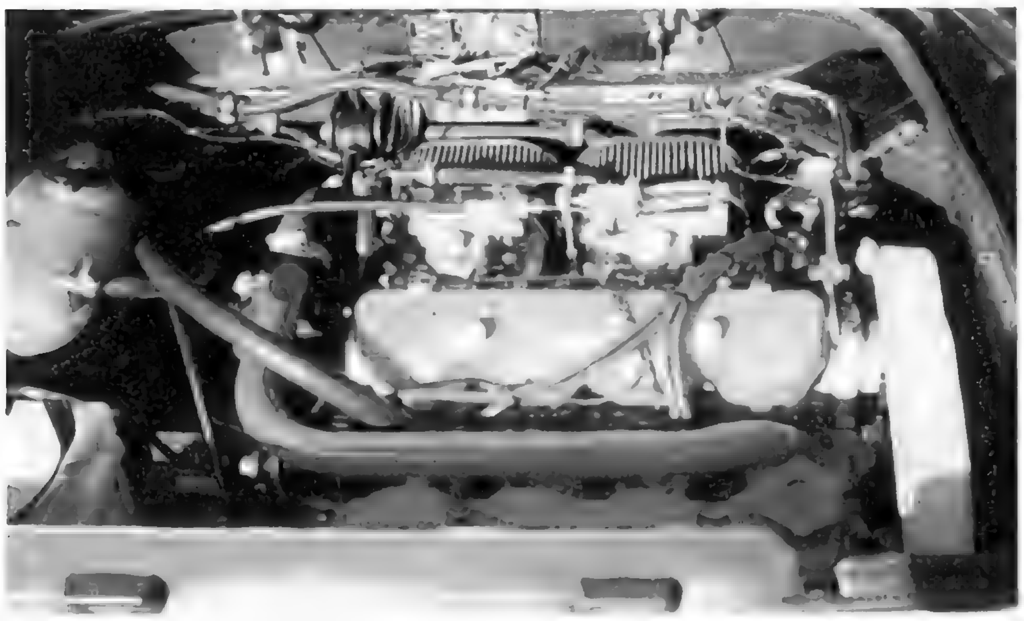
18/59: Meticulous build-up of Martin Gallacher's 130TC rear-wheel drive Westfield conversion. Spec includes GC-prepared high-compression block assembly (10:1 CR), St III cams, ported 130 head (43½/36 valves) and 45 carbs (38 chokes). Output is in region of 178bhp @ 7000rpm.

18/60: Ivan Mostyn-Scott poses happily with 130 TC shortly after conversion to 45s using GC adaptor plates. Pleased? Ivan's face says it all.



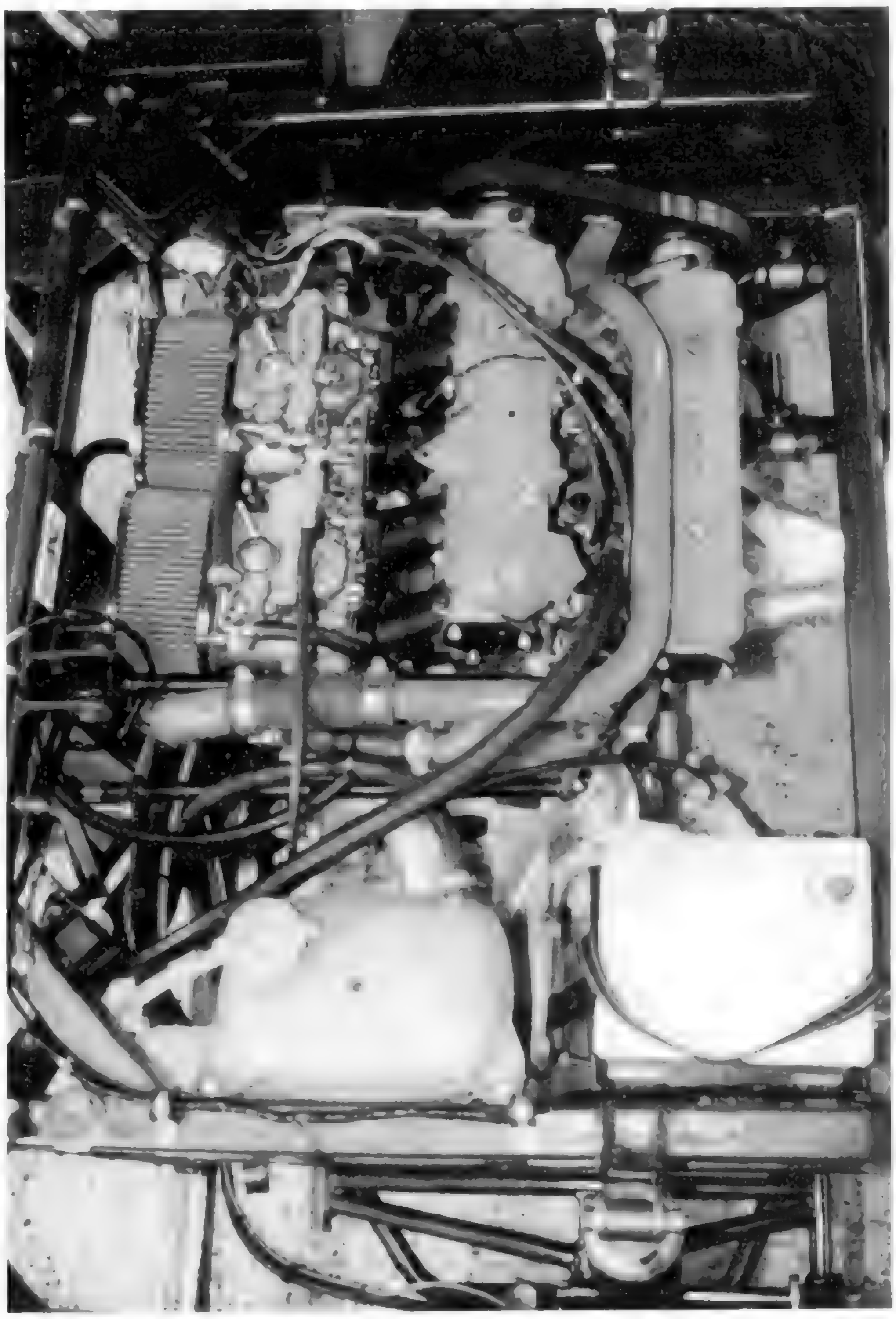
18/61: David Light from Basingstoke built this JH Classics 246 replica equipped with a GC 2l Lancia St II. Car is breathtakingly quick: St III rally cams, ported/blueprinted head, 10:1 CR (unleaded seats), 42½/37 race valves, 45 DCOEs (see Case History No 8).

18/62: "One day ..." Kit Patana from Thailand (who trained at GCT) with Dave Light's 246 replica at GCT in '91.



18/63: Proximity of engine to bulkhead does not prevent sidedraught carbs from being fitted. (Author's note: Flexibility of engine was extraordinary. During a test drive I commented to Dave Light on how tractable the car was at around 12mph in second gear, only to be told that we were in fact in third!)

18/64: Engine bay of Dave Light's beautiful 246 replica. Exhaust is later 4-2-1 type; early 4-2 type gave problems with short primary pipes and split secondaries.



18/65: Versatile nature of Lancia 2l powerplant is evident in this shot of Light's car. When twin filters are used (these are K&N) balance is affected by vacuum in inlet manifold which moves carbs relative to each other. One-piece backplate gives better result, holding carbs in line. Shot shows early 4-2 exhaust.

18/66: Thai engineer Kit Patana at Bangkok race circuit with his beautifully restored 124 Special T. Likeness to Lada is unmistakable. No helmet, racesuit, cage or harness required!

18/67: Rod Bennett rallying his 1600 Guy Croft Delta on the Vauxhall Sport rally in North Wales, March '90 – DNF (Did not finish – driveshaft sheared!). (Photo Speedsports, Ruthin)

18/68: Superfast Delta 1600 rally car belonging to Rod Bennett. Dyno-tested on Go Power rig in February '90, developed 164bhp (corrected) at 8000rpm, 124lbf ft torque at 5500rpm (with not a lot below!). With straight-cut gearset and 7¼" race (twin-plate) clutch, sounded terrific on stages – "like a gas turbine" commented one marshal! Rev-limiter was set at 9600rpm and it was all used! Engine spec comprised 45s (36 choke), St III rally cams, 44/37 valves, 18" 4-2-1 manifold, forged pistons, 10.8:1 CR, fully heat-treated crank, steel flywheel, dry sump. GC engine No 78 finally met its demise when stones caused oil pump drive belt failure – a major problem on rally cars.

18/69: Vauxhall Sport rally, March '90. Note serious front lighting! Best result with car was Imber Stages in 1990, 3rd in class, (against 2l 16v machinery); at one point Rod and Rob (navigator) roared past a 16v Integrale – probably the high spot of the day! Although Imber is a tarmac rally, car arrived in 'forest' trim – Colway tyres, forest suspension and gearing, and with defective brake bias valve into bargain. Only superb driving and phenomenal flat-out performance of Delta engine led to such an amazing result! (Photos 18/69 to 18/75 Speedsports, Ruthin)

18/70: When Rod Bennett had graduated to HUL 774W after selling his Fram Welsh class-winning 128, navigator Rob Langley commented: "I can't keep up, it's too fast". What he didn't know was that two years later…

18/71: …Rod had his eye on rallying a full-spec Gp A 8v Integrale!

18/72: A proper rally car being used for its design purpose! Rod Bennett at speed in the forest on route of previous RAC rally.

18/73: Rod Bennett, Manx International May '93. Full-spec Gp A car: engine built at GCT with 28lbf/in^2 boost (copper head gasket and high-strength bolts, forged pistons, fully ported/blueprinted, Abarth Evolution cams, developing around 340bhp. Rod comments "The Manx is the nearest thing to forest on tarmac!" Huge brakes needed with this sort of performance. Discs are about 1ft in diameter.

18/74: Manx National, April '93, Rod Bennett and Rob Langley achieved very commendable 60th overall (out of about 180) with 5th in class. A major achievement for a minor team, especially in view of high cost of running such a car.

18/75: "The Manx is a love-hate relationship", said Rod afterwards. "You either love it or hate it – I loved it, and I can't wait to do it with the NEW 16v car!" Note full Gp A Abarth rollcage.

18/76: Rod Bennett tackles watersplash on '93 Manx National. Other major hazard was dry-stone walls. Navigator Rob Langley was horrified to look up at one point and see them flashing past at 120mph inches from his window!

18/77: 1994 RAC Special Stage 1 – Carden Park – still clean! Rod Bennett and Rob Langley have come a long way since their 1988 Fram Walsh 128 win. The first, and hopefully not last, Guy Croft engine to compete on RAC.

18/78: SS 3 – Chatsworth, first water splash! Note roof vents added since Manx. Nearside front tyre went flat on startline; Rod drove 6½ miles – only dropped 1½ min! Twin differentials were fitted for RAC due to handling problems.

18/79: SS 4 – Clumber Park as night falls. At service areas, crew were mobbed by rally enthusiasts demanding to see 'a real rally car' and requesting autographs! Despite problems of financing such a costly enterprise, this private team demolished dozens of entries from more expensively equipped teams until, on the last stage, broken hub studs stranded them within minutes of finish.

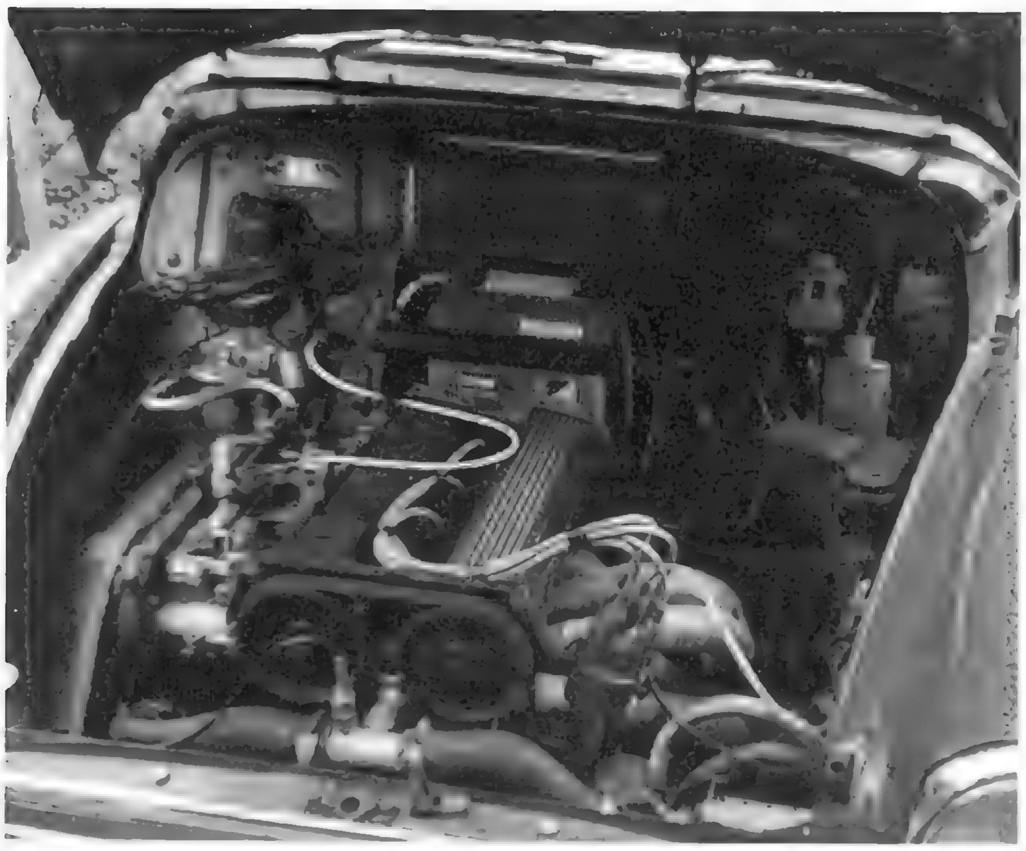
18/80: SS 8 – Hamsterley. Rod commented: "By this ford I'd overtaken one car and caught another". Sadly, drowned electrics and one-cylinder engine lost them 10 minutes before it dried out. Rod has now set his sights on a 16v engine: homologation regs allow 8v shell to be upgraded.

18/81: Mint-condition US-spec Pininfarina Azzurra (124 DS) imported by Tony Dark of Beaconsfield, Bucks, in '93. Beautiful car let down by soggy performance of US engine (albeit fuel-injected); 8:1 CR and US cams led Tony to specify engine rebuild at GCT.

18/82: It's quite true, as Phil Ward wrote in his book Fiat and Lancia Twin-Cams – you can raise and lower the hood with one hand! This particular car is much admired – styling is a Pininfarina classic.

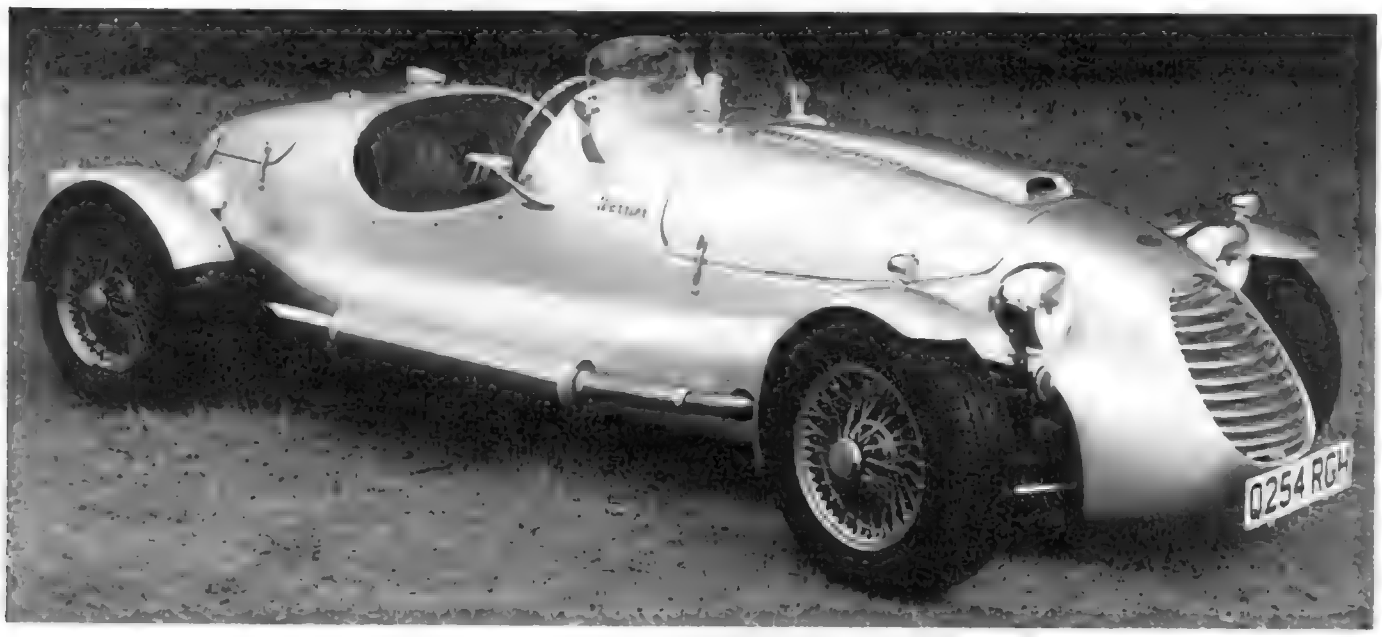
18/83: Tony Dark's rebuilt engine, GC No 202, featured 10:1 CR, fully ported/blueprinted head, lightened flywheel, 131 Sport cams. No mods were required to fuel-injection system – engine ran perfectly with standard set-up. Mid-range torque now very significantly higher!

18/84: Roger Visocki, of New Jersey, USA, competing with highly modified Spider in SCCA race. (Photo D & R Auto Service, NJ)



18/85: Martin Baker's 'wolf in sheep's clothing'! Thema Turbo 16v head grafted to 131 2l block with GCT cams and 16v ie pistons, develops 190bhp @ 7200rpm at wheels! Note ITG filter and airbox – feature often omitted but well worth effort as high bonnet temperature robs power. Carbs are 45 Dellorto (40ch) fitting: No 6 emulsion tube, 200 needle valve, 60.9 idle jet, 140 air corrector (for top-end enrichment), 150 main jet, drivable below 5000rpm but will not take full throttle. Characteristic of a 16v with its large ports and low inlet velocity at low speed.

18/86: Alan Hooper (owner of HDS cars of London) with his Type 48 Warrior, based on a 1948 Grand Prix Ferrari, at old banked circuit at historic Brooklands. Car is fitted with 140 TC engine adapted for rear-wheel drive and with 45 carbs.

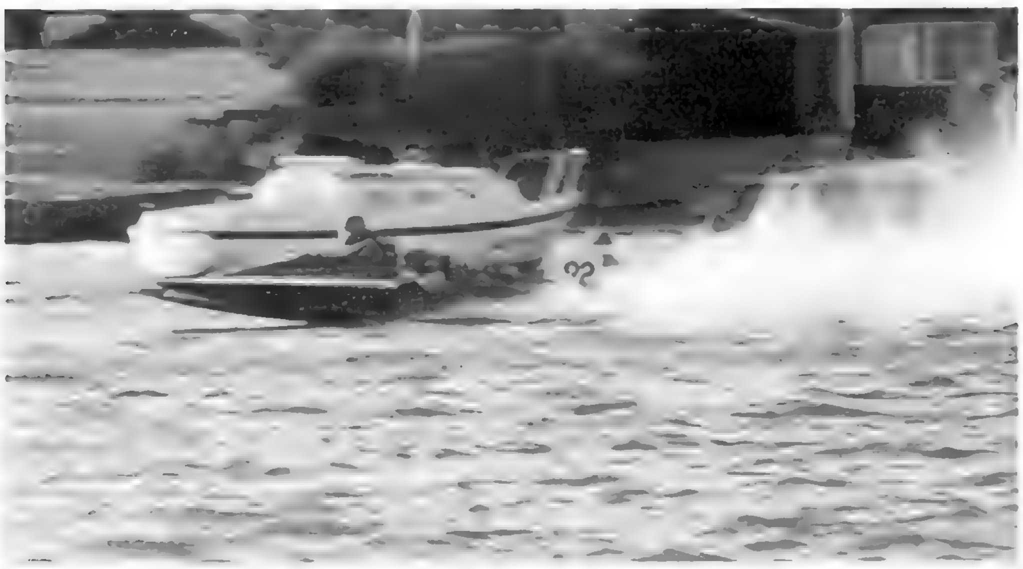


18/87: Entirely designed and built by Hooper, car is constructed to the highest standards. It drives and handles superbly and is one of the most responsive 'component' cars driven by the author. Car represented a bold attack on stranglehold of 'Seven-type' cars on kitcar market.

18/88: Two into Uno will go, as illustrated by this shot of a modified Lancia TC in a Fiat Uno.

18/89: Mk I race boat! This three-point F2000 hydroplane owned by Neil Allen of Lowestoft was raced with 1800 Fiat built by GCT during '90 and '91. Seen at speed on practice course at Lake Windermere, Cumbria, during 'Record Week'. Despite atrocious conditions, boat achieved two-way average of 97mph (British record 101mph)!

18/90: Hydro racing at Stewartby gravel pit. Three-point suspension evident – propshaft just visible; suspension comprises prop and tips of sponsons. At low speed, hydro behaves like displacement hull; as forward speed increases, air cushion in tunnel between sponsons lifts craft clear of water.

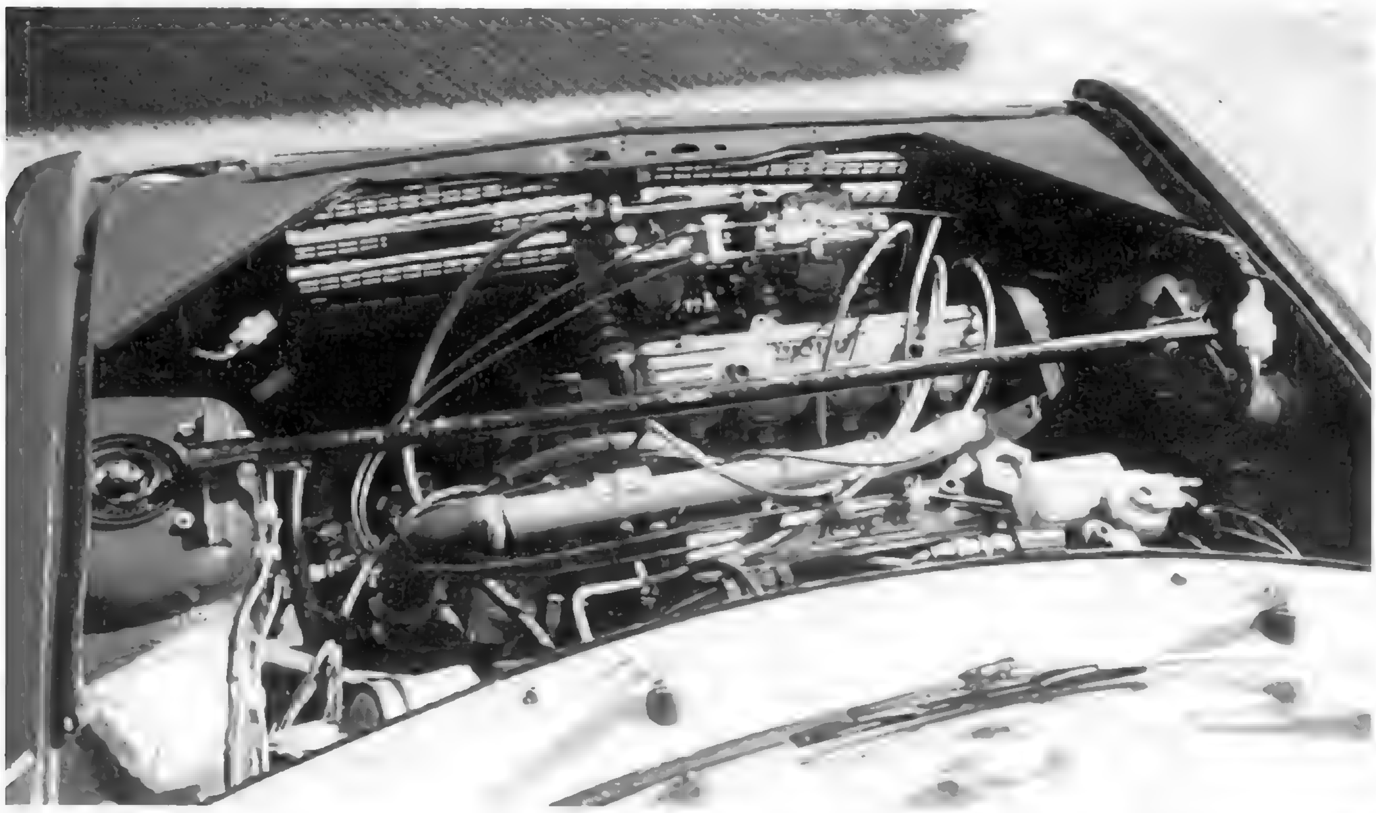


18/91: Oulton Broad, '90. Neil Allen powers his craft to another victory. Despite race handicap which prevented Allen joining course until slower craft had completed two laps, Fiat 1800 hydro was easily capable of entering 10-lap race and winning by over half a lap! Engine drove propshaft off crank nose; because of engine rotation/propeller availability, drive off flywheel end was impossible; no gearbox to reverse drive was permitted. Early problems of broken crank due to torsional stress were overcome by heat-treating crank and reducing flywheel weight to bare minimum. Boat won Dutch Grand Prix in 1989.

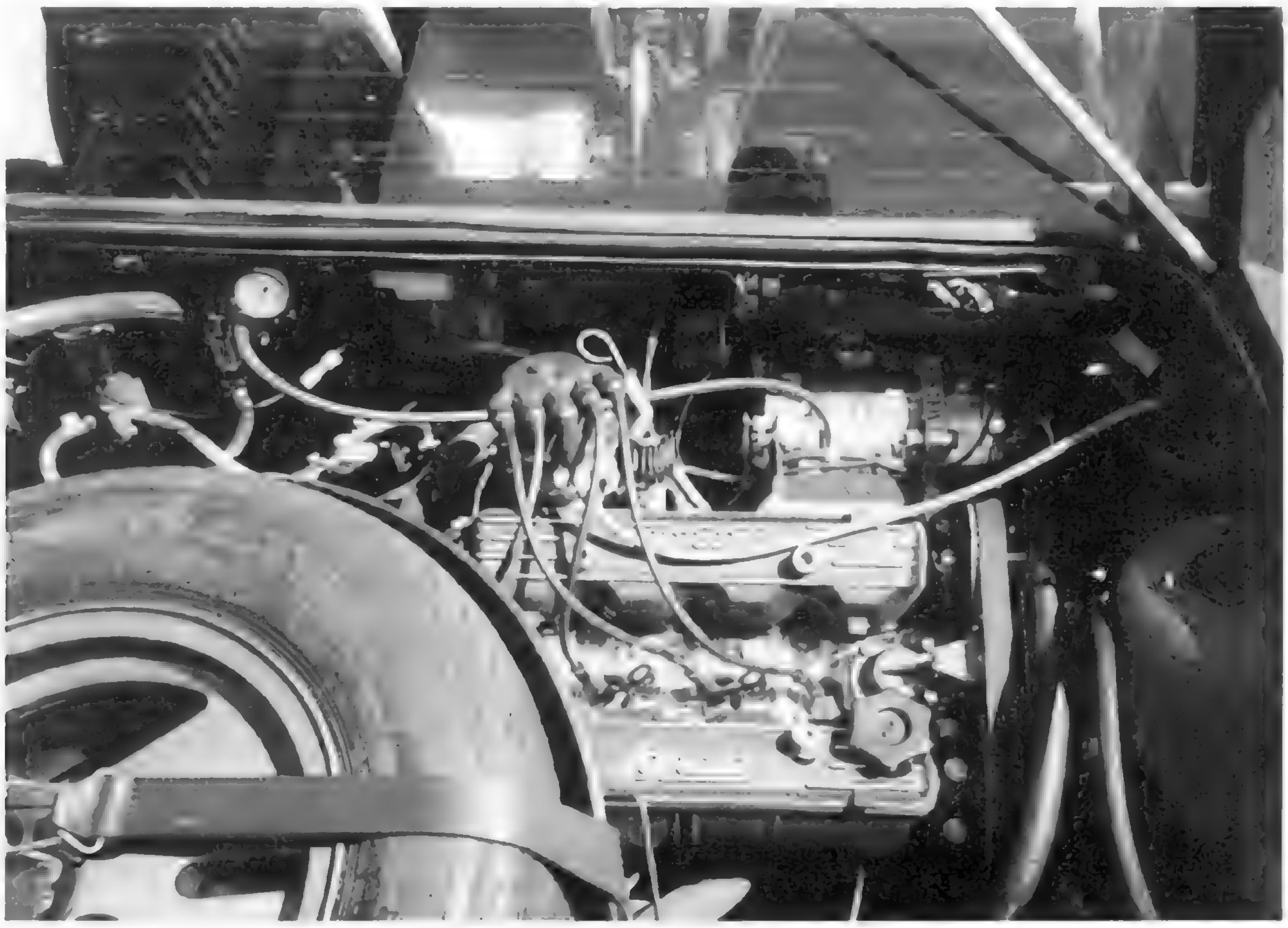
18/92: Mk II boat; 18" longer, wider, with more powerful GC 1800 engine. Broke all-time Oulton Broad lap record in '91, totally outclassing the Alfa and Imp-powered opposition.

18/93: Giant killer! Best-of-both-worlds Mincia, designed and built by Robin Cooke of Eastbourne, incorporates classic Issigonis body with classic Lancia 1600 TC engine. Radiator from Rover V8 is in boot – nowhere else to put it!

18/94: Initially fitted with a twin-40 standard 1600 Lancia engine, car caused sensation when first shown and was featured in several magazines including Cars & Car Conversions.

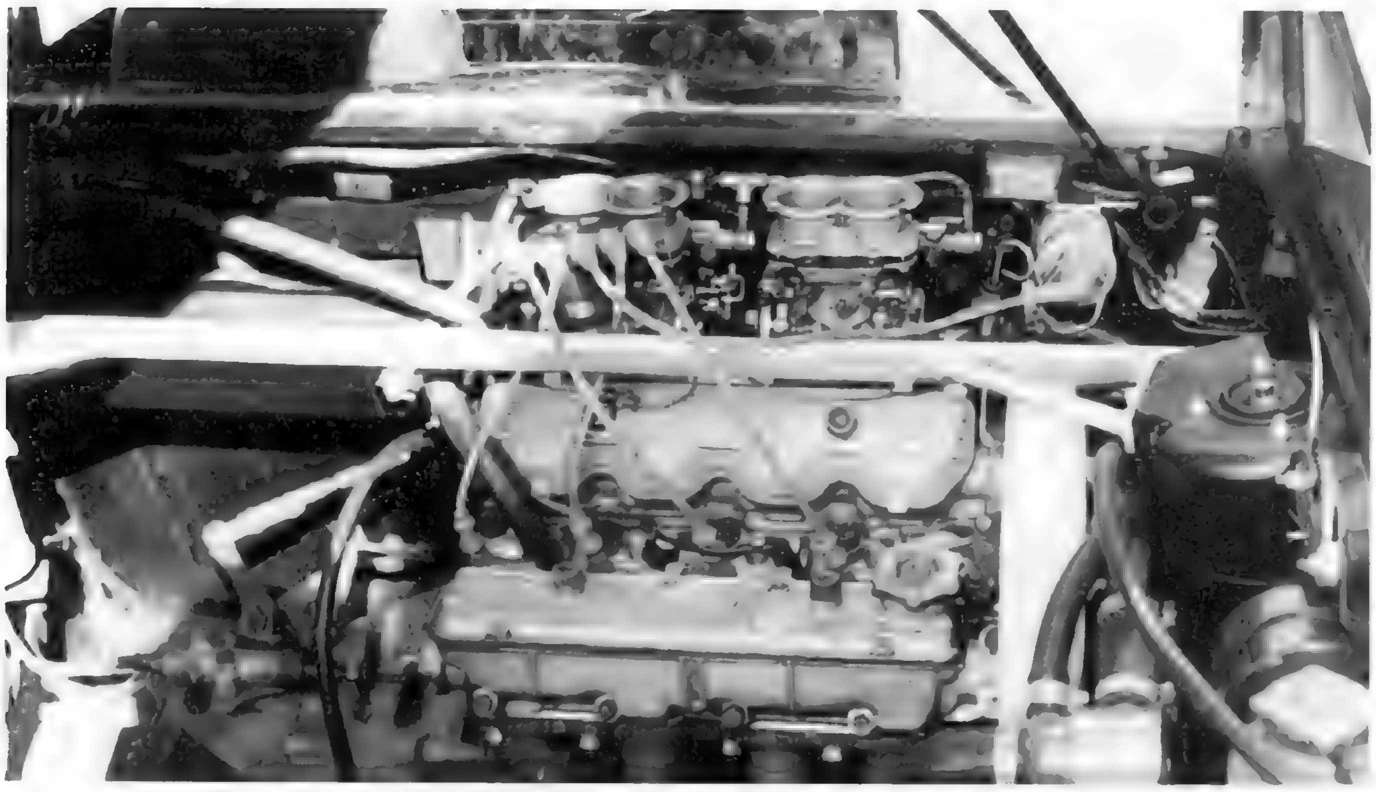


18/95: Complete Lancia subframe grafted to Mini shell. Latest engine is GC St II developing around 145bhp @ 8000rpm. When Cooke first took it out on test he reported: "Quite tractable at 2, 3, 3500, 4000, then suddenly at 4500 it took off like a scalded cat!" Car won 1st in class in '94 Brighton speed trial and gained 3rd in class at Valence Hill Climb – no mean achievement for a car with a standard Lancia Beta gearbox.

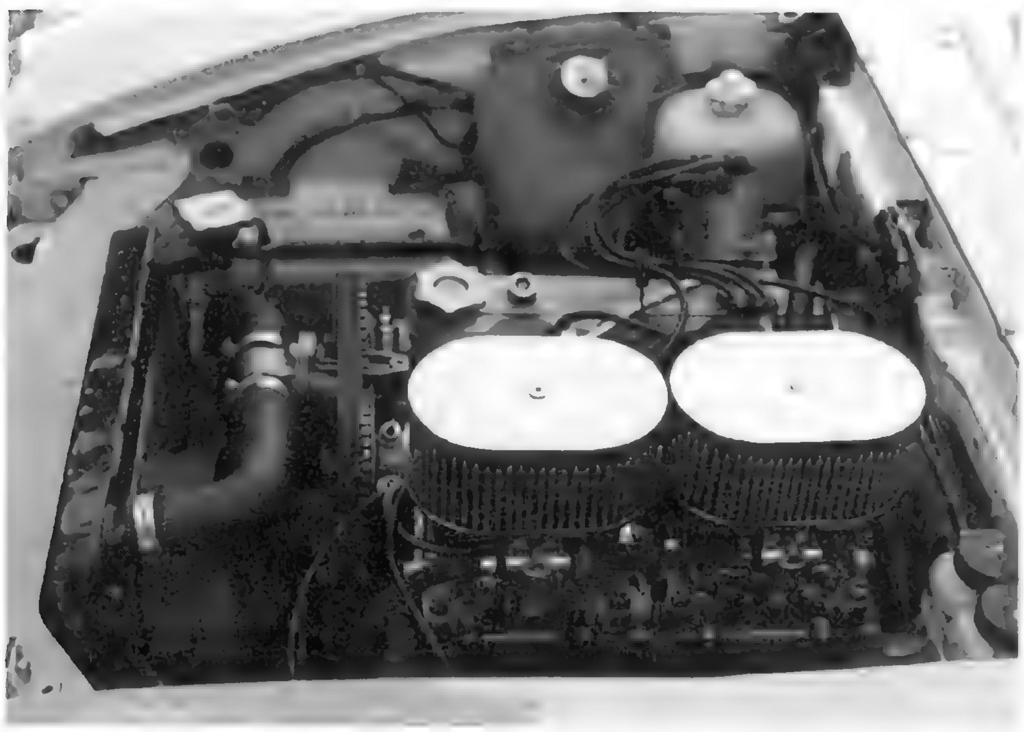


18/96: Tom McGaffigan's neatly installed Volumex-blown Monte Carlo engine. Blower mountings had to be designed to allow fitment under bulkhead. Note Monte Carlo inlet-driven distributor – only ever used on that model.

18/97: Beautiful Monte Carlo formerly owned by Dave Sharpley featured GC engine, ported/blueprinted, standard cams, 44 DCNF carbs, 9.6:1 CR, produced around 150bhp @ 6300rpm.

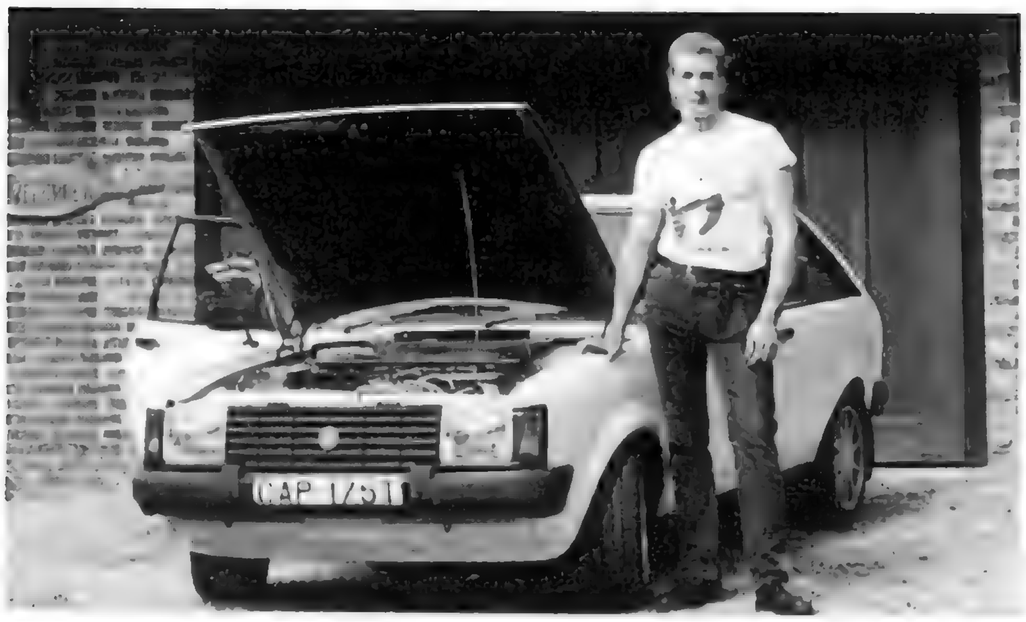


18/98: Dave Sharpley's Monte Carlo engine. Became a Guy Croft customer when reground cams fitted 'somewhere else' proved hopeless – chronic fuel 'stand-off' above trumpets, loud roaring and minimal torque.

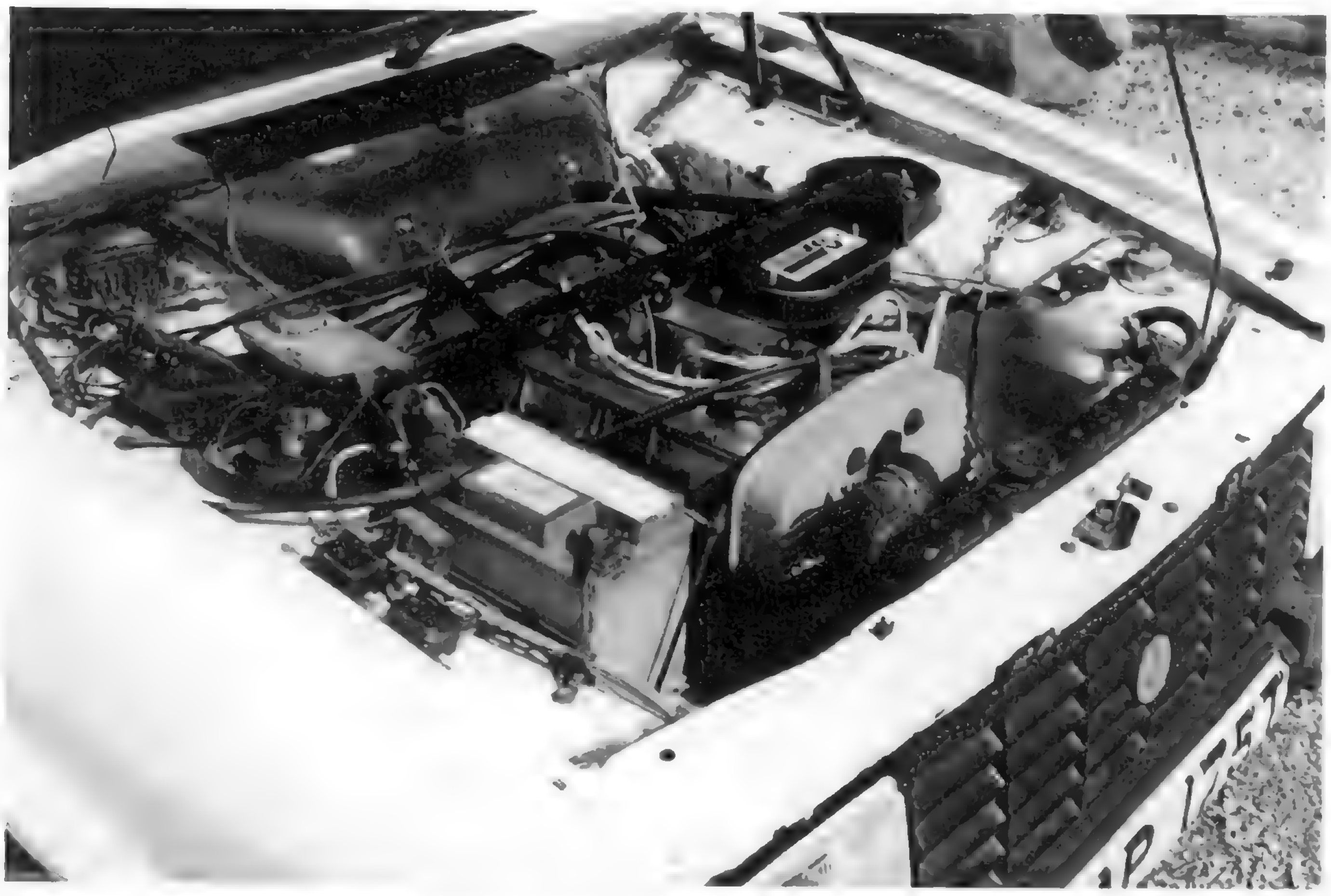


18/99: Author's 124 CSA 1800 in original homologated trim except for addition of pair of K&N filters. Car was bought from a director of Lancia in 1984 and driven back from Turin on long weekend leave from military service!

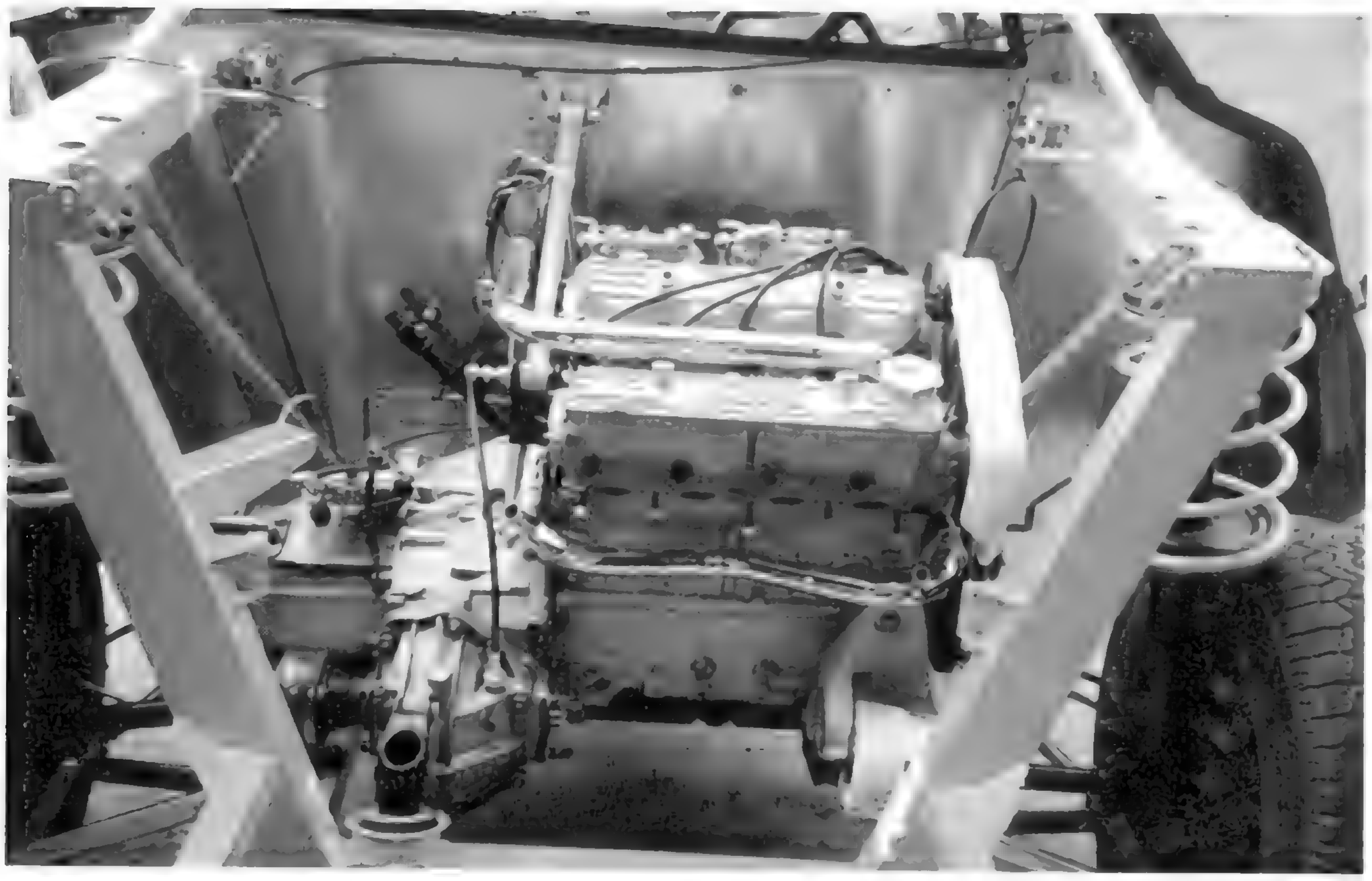
18/100: Troy Dunlop's GC-powered Monte Carlo set many early records in Italian Intermarque Challenge. Forged pistons, 46/40 valves, St IV cams, 48 carbs (42 chokes), dry-sump lubrication netted 196bhp. Constant dicing with Duckhams QXR – Fast Car Monte Carlo during 87/88 made exciting viewing. Mid-engined set-up made it a particularly quick car off-the-line. 4-1 exhaust was fabricated by Mike 'The Pipe' Randall, of Wallington, Surrey; one of top men in business.



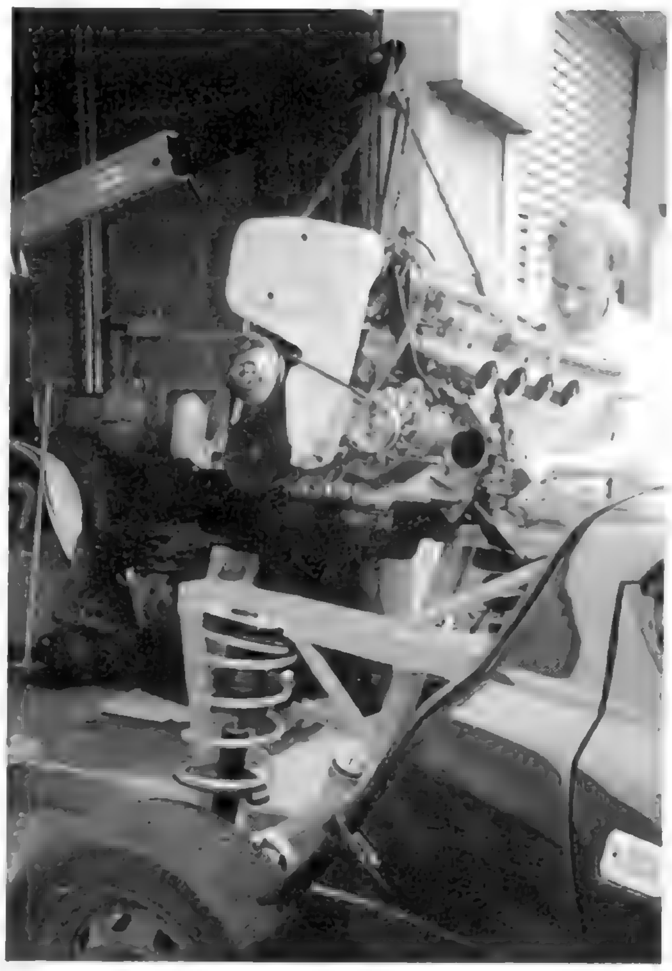
18/101: Adam Laszkiewicz with his home-built Talbot Sunbeam-2l Fiat.



18/102, 18/103: Talbot Sunbeam is ideal conversion for 2l Fiat as these two shots show. Engine features GC-ported/blueprinted head, standard cams, 9.6:1 CR block; around 125bhp.



18/104: Engine bay of Phil Jordan's Transformer Stratos replica with GC engine No 53, 'fast road' 2l Beta, installed. Ported/blueprinted head, 45 carbs (36 chokes), 2014cc, 9.6:1 CR, standard valves/cams, gave 155bhp @ 6300rpm. Clever design of layout made for very easy installation.



18/105: Enthusiastic Phil Jordan carefully aligns motor at GCT. Note alternator upper bracket – standard Beta item minus rear strut – proved quite satisfactory. Thermostatic sandwich plate and Volumex filter used. Beta inlet manifold gives correct carb position even though engine has 20° tilt.

18/106: Some five years after initial start-up, Jordan's engine still going strong. Here, with wife Catherine as navigator, he puts power down on '94 Classic Marathon. By now engine has GC race sump! Broad spread of torque (engine only loses about 8lbf ft between 3800 and 6000rpm) makes car as quick as many other competitors with more power. Historic shot of faithful replica being used for its proper purpose!

18/107: Engineer Leif Bengttson of Sweden built this immaculate Bentec 7 for Swedish Sports Car racing run by MG/Gothenberg and Stockholm Car Clubs.

18/108: Mild oversteer on this aluminium-bodied Super Sette as Leif enters a turn during club racing. Compression of offside suspension indicates car is not travelling slowly! Gearbox is 5-speed Sierra with Quaife gearset. Tyres are Yokohama 008R which Bengttson rates very highly for road/race.

18/109: Road Sport Class A. Kinnekulle circuit, central Sweden, '92. Bengttson beat 944 Turbo and Caterham with his Bentec 7, at that time fitted with Ritmo 125 TC engine.

18/110: Second race at Rudskogen, near Oslo in Norway, '92. Leif (right) takes 3rd place against Lotus 7 (1st) and Caterham (2nd). Race classes are cleverly structured: cars compete according to power/weight ratio tables drawn up by organizers.

18/111: Swedish hillclimb event in '92. Bentec 7 cars with two Caterhams. Leif clinched fastest time of the day overall, including Modsports.

18/112: John Camm racing his 131 Sport 2l during Italian Challenge in 1987. Engine was ported, blueprinted with twin 45s and often quicker than cars with peakier units due to flat torque curve. Sadly written-off when defective front dampers caused Camm to lose control of car at Brands Hatch. Heavy nearside front-end meeting with Armco caused floor panels and wings to open up like a sardine tin – highlighting essential need for seam welding on any competition car. Bonnet folded and scythed backwards – only prevented from shearing through windscreen by secure bonnet locking pins. Engine developed about 145bhp on 9:1 CR, but was easily outclassed by more powerful GCT race Monte Carlos in 1800–2l modified class. However, car proved useful test-bed for early Guy Croft Fast Road 2l engine.

18/113: Interior of Mick Wood's fearsome Gp 4 124 Abarth Spider. Still sporting its original Abarth 068 cams, engine mated to dog-clutch Colotti box provides phenomenal acceleration. Car still features (and wins!) in historic rallies.

18/114: Mike Wood's car on visit to GCT in 1993. Very few of these sensational cars now remain; most were wrecked by works teams in '70s. Unique Gp 4 features are original front and rear Abarth suspension, alloy doors and sills, glassfibre bonnet and boot.

18/115: Superfast Lada TC rally car owned by solicitor Anthony Fausset, of Houghton, Cambs. Engine is St II GC transplanted from Tony's 131 Sport!

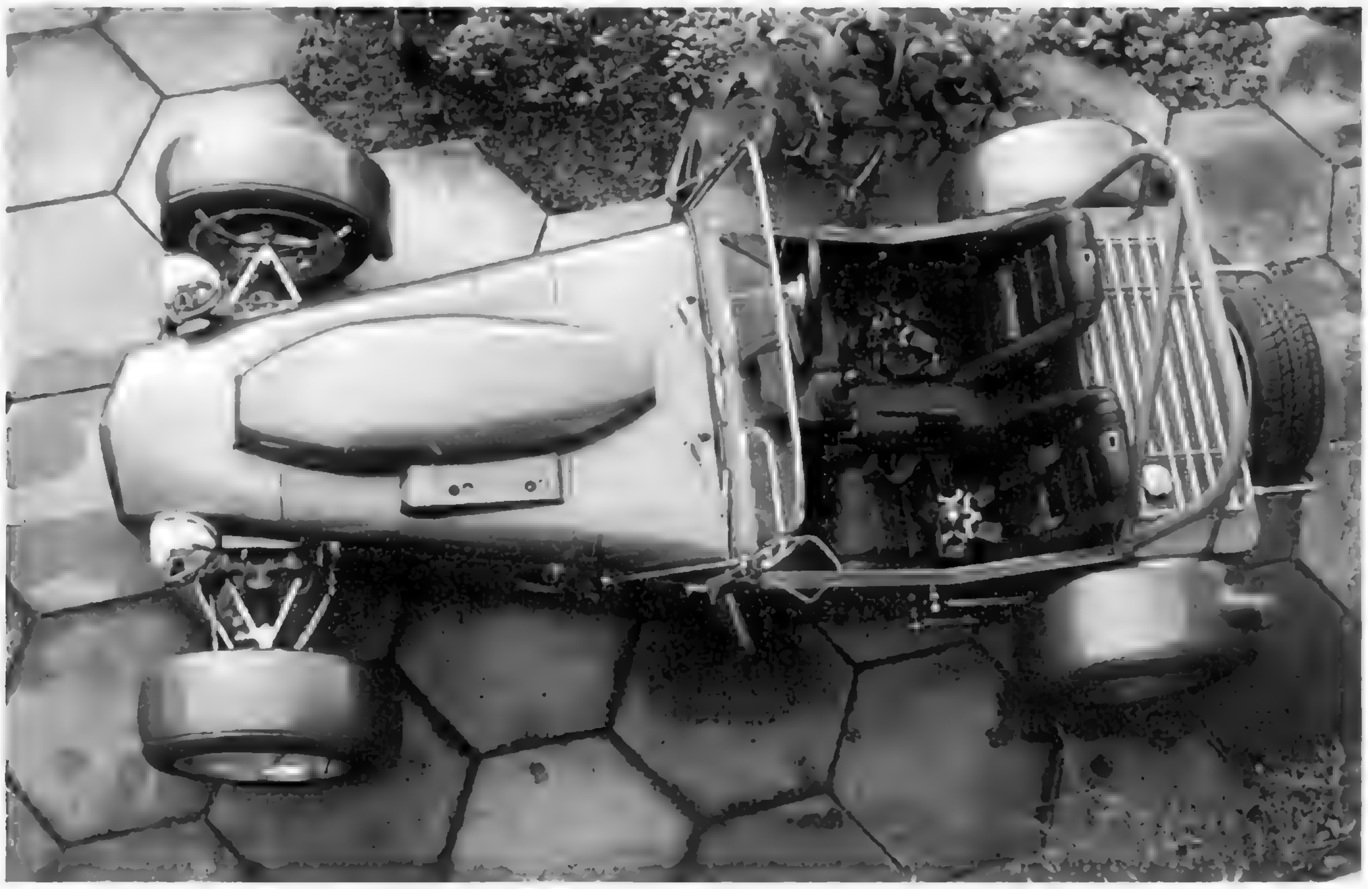
18/116: This East European derivative of 124 ST handles superbly and would make excellent clubman rally car. Driven by the author, engine response was instantaneous! Reminiscent of Seat works cars of '80s.

18/117: Marios Lourides, of South Ruislip, Middx, owns this attractive (and now rare) example of Beta Spider 2l.

18/118: Nearside view of engine bay. Air filter located close to radiator saps power and upsets jetting of carbs; alloy baffle plate or airbox would be better. Choke connection is redundant really. A well set-up TC should start after three or four pumps on throttle. Note that Beta engines were all inclined back 20°. Bosch electronic ignition amplifier visible on rear of offside inner wing.

18/119: Marious Lourides' Beta 2l Spider. Note there is enough room for Bosch/Marelli alternator even with filter fitted. Owner's intention is to restore car. His verdict on carb conversion: "Brilliant. I love it."

18/120: Liam and George Scott from County Down built this superb Autotest car back in 1969; design was years ahead of its time. Original engine was 1650cc Ford crossflow. Features Cortina rear axle, Triumph Herald uprights and custom-made wishbones.

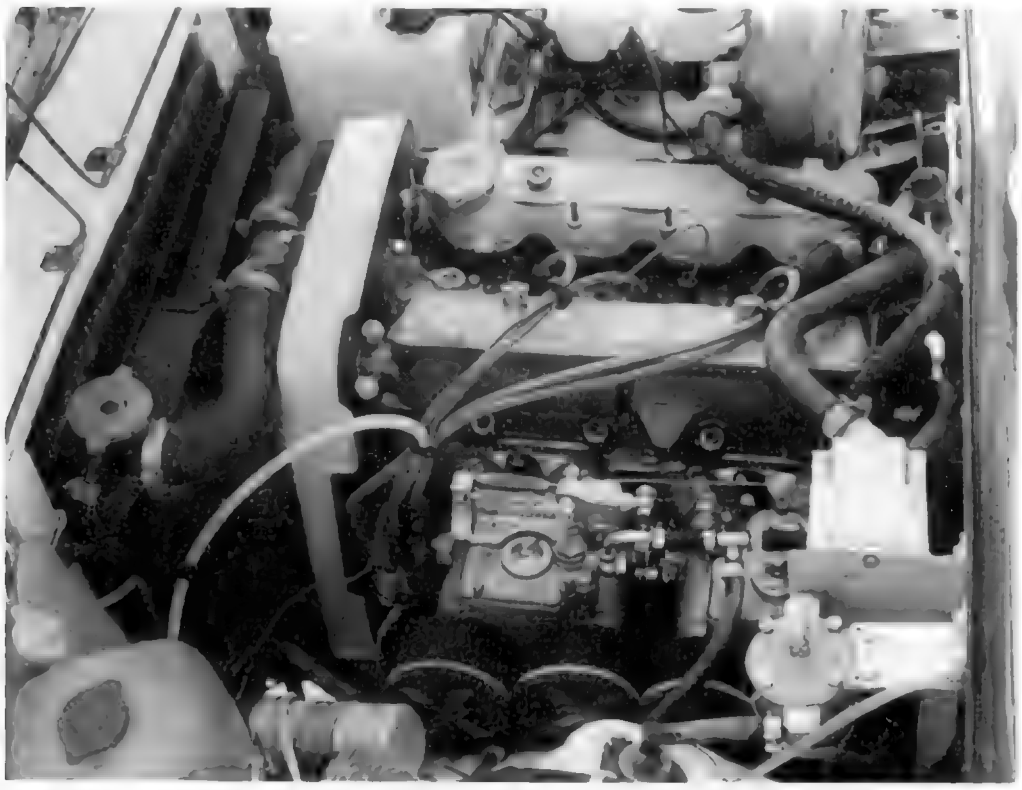


18/121: During winter '94 car was rebuilt with 2l Fiat TC "with advice from GCT". Car looks ready to take off on its own in this aerial shot!)

18/122: Detailing is flawless. Attractive, period, Cortina 1600E instruments give comprehensive readout of engine functions. Bonnet bulge conceals heater connections.

18/123: Rear-three-quarter view illustrates funtional nature of purpose-built car. Design criteria: car must go forwards, backwards, round corners, sideways – quickly! Autotest around artificial obstacle course requires maximum agility from car and crew. Future use will include odd foray into classic rally (26-year-old car!) and hillclimb.

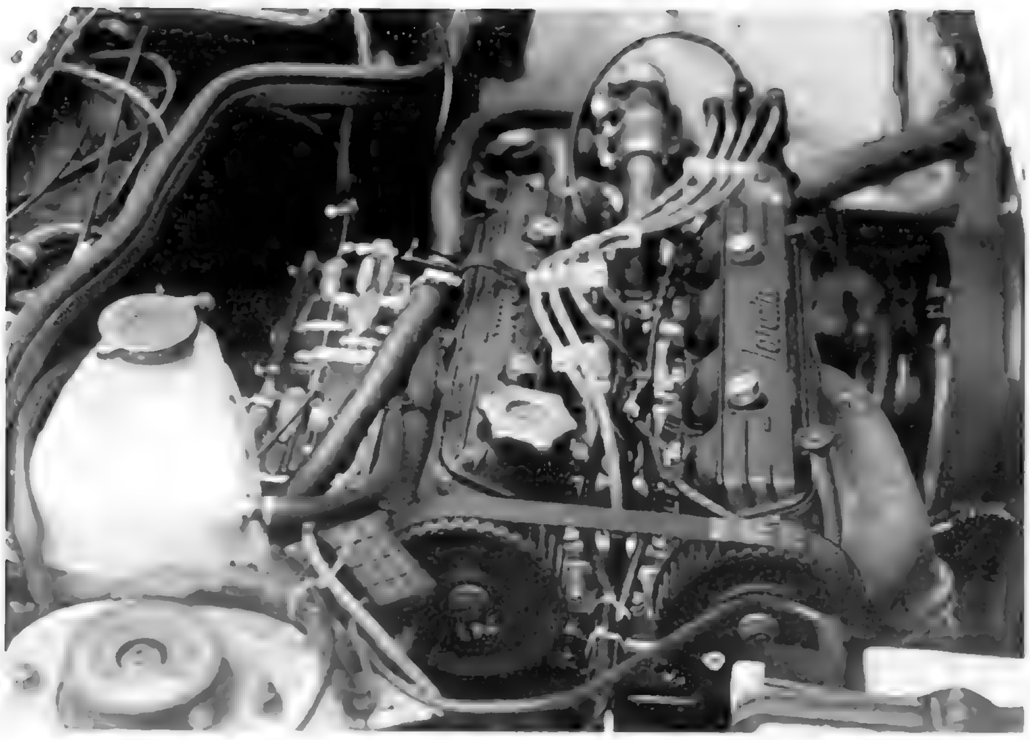
18/124: Next time you pull alongside a white FSO pickup, look carefully, especially if it has twin sidepipes…



18/125: …it could be equipped with a 145bhp 2l Fiat (GC ported/blueprinted head, sidedraught 45s) and owned by genial haulage contractor Roger Smith, of Hoo, Kent.

18/126: Engine bay of Roger Smith's car – clean, tidy and everything exactly where it should be. Super conversion.

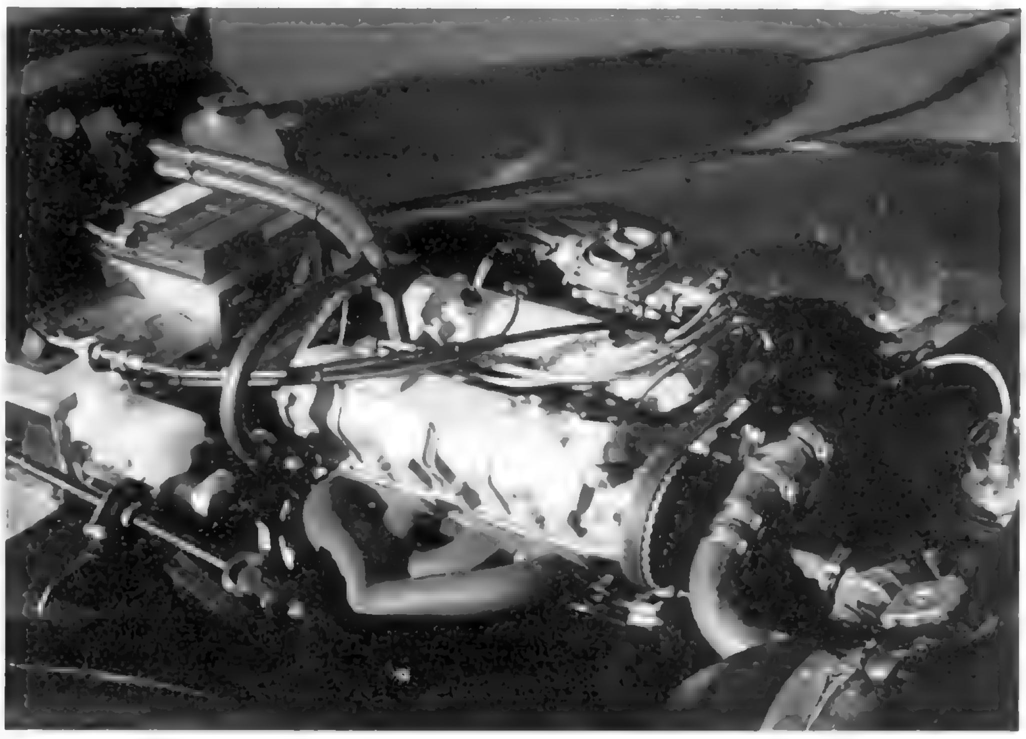
18/127: Apart from general 'gopher' jobs expected of a pickup working for a haulage company, Fiat TC/FSO is also periodically required to embarrass more exotic machinery at traffic lights. In this role, apparently, it succeeds admirably, having 0–60mph in 8.0sec; top speed not sure as speedo runs out at 110mph.



18/128: Better known for his work with veteran Lancia engines (Fulvia, etc), engineer Peter Gerrish built this 2l conversion for a Delta. Heater fan duct has been modified to accept carbs on a straight-shot GC manifold. Note reversed-port (late) head.

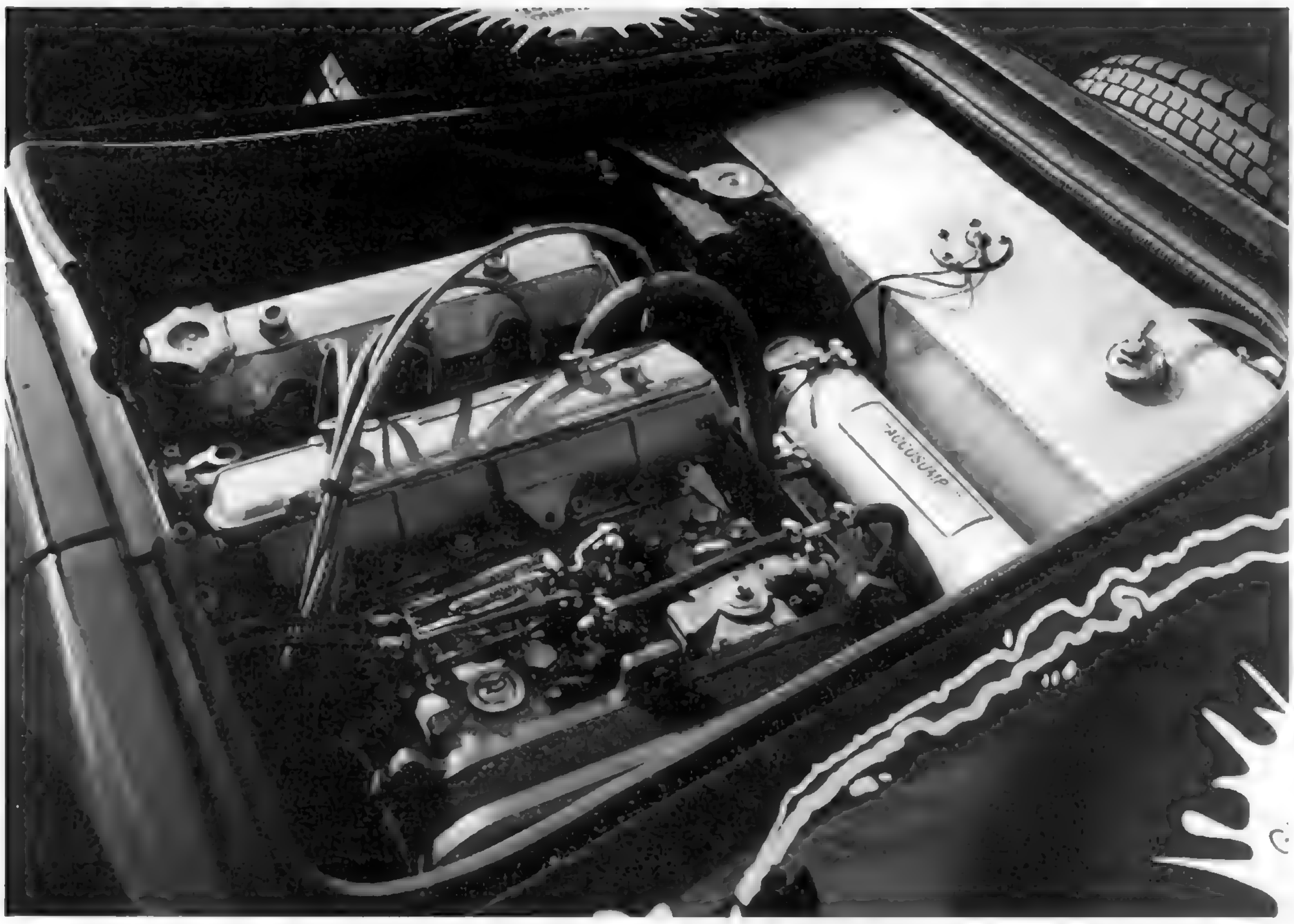
18/129: In May '92 Gareth Jones, from Shrewsbury, bought a 120,000-mile W-registered 131 1600 TC Estate…

18/130: …this part of it soon being marked destination Westfield!



18/131: Jones writes: "I rebuilt the engine to almost standard spec. The only excessive wear was to the piston bore, and +0.4 pistons solved that minor problem; the crank was checked, polished and refitted, as were the cams. The valves and seats were machined and new guides fitted…

18/132: …although it has not been tuned yet, it runs and sounds great!" Adoption of a TC? Accessibility to such a simple yet powerful powerplant is main reason for choice by so many kitcar owners.



18/133: Director of Midtec Sports Cars, Terry Mulloy, adopted a GC St II engine for his 'demo car' in 1994. (For full spec, see Case History No 1.) Performance is blistering with 182bhp @ 7000rpm. Carbs are DCOE 45 (38 choke) with Weber linkage, filter is one-piece ITG. Note Accusump: essential on an engine frequently used at race circuits. Radiator is in front of car; heat exchanger cools oil. Engine is mated to Renault 21 Turbo box.

18/134: Car was extensively featured in Kit Cars International and Which Kit magazines. Styling rather unusual, but concept can't be faulted and handling is superb. Midtec director Terry Mulloy at wheel on Castle Combe race circuit at one of many kitcar 'Action Days'.

18/135: Functional styling? Compared with the excavator, 'futuristic' seems a better description.

18/136: Gerry Hawkridge owns this superb HF 2000...

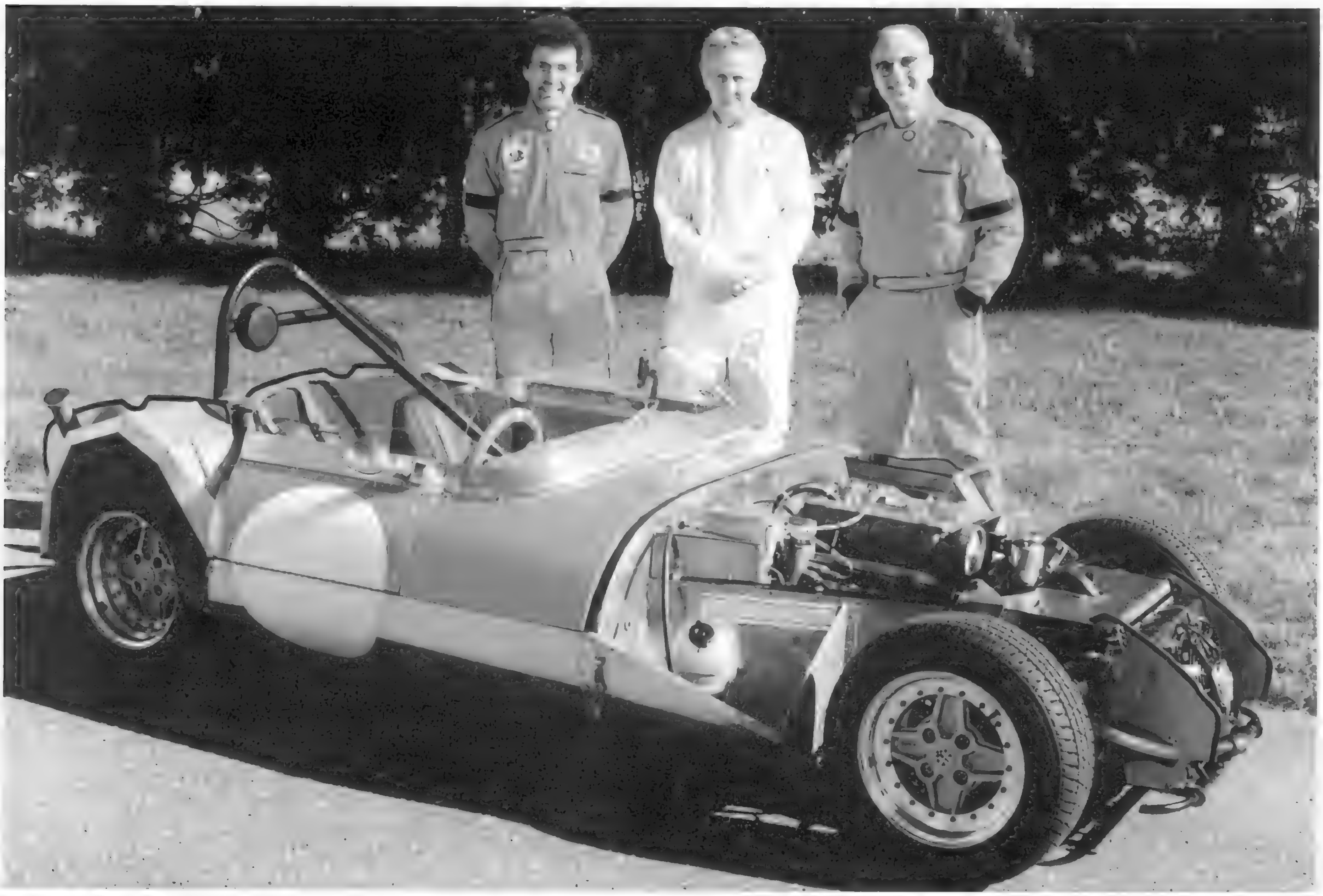
18/137: ...fitted with a GC-prepared Volumex unit. Car was featured in Kit Cars International in early '94; these photos were taken by editor Ian Hyne – one of the most knowledgeable people in the business. Engine has 44 inlet valves, Beta ie inlet cam, modified pistons, fully ported/ blueprinted, balanced. On 40 DCNF produces around 170lbf ft of torque (rolling-road figure). Ignition pack is standard Marelli (Volumex). Ducting would improve airflow to carb – high-temperature power loss must be around 5–10bhp.

18/138: Massive tyres cope with the torque of Gerry Hawkridge's immaculate HF 2000.

18/139, 18/140: [Author's note: I have included this data, supplied by Dave Massey, to illustrate the attention to detail that has been a feature of the preparation of this outstanding car.]

WESTFIELD ELEVEN
Owned, built and run by John Hostler, Ted Cox and Dave Massey. Campaigned in '92 and '93 AEMC Sprint Championships with 98bhp MG Midget 1275cc A-series engine. The car has been modified during 1994/5 to the following specification, including the fitment of a Fiat 1600 Twin-Cam power unit, in order to compete in the '95 AEMC Sprint Championship in the roadgoing Caterham/Westfield class, up to 1700cc.

CAR SPECIFICATION	
Model	Westfield Eleven
Serial no	22
Date of build	1983
Chassis	Square-section steel tube fabrication
Chassis modifications	LH side of engine bay lowered and strengthened to accommodate carburettor cold air box
	Front suspension pick-up points for top wishbone (previously single-link/anti-roll bar)
	Front anti-roll bar mountings
	Modified crossmember beneath gearbox
	New gearbox mounting crossmember
Inner body	Rivetted aluminium sheet
Outer body	GFRP. Lift-off front and rear body sections
Clutch	AP Racing friction plate and cover, standard release bearing
	Hydraulic release: Midget master cylinder, Toyota Supra slave cylinder, modified Fiat release arm
Gearbox	5-speed Fiat Supermirafiori 1600
	Output flange modified to accept u/j propshaft
	Shortened 131 Sport gearchange extension with selectable lock-out of 5th and reverse
	Clutch release arm re-orientated for top exit and converted from cable to hydraulic release
	Redline synthetic oil
Rear axle	MG Midget wire wheel (narrow) case with shortened steel wheel halfshafts
	3.72:1 final-drive ratio (Riley 1.5)
	Pawl-type limited-slip differential
	Shortened Marina propshaft
Front suspension	MG Midget uprights and hubs
	Fabricated steel tube upper and lower wishbones, adjustable anti-roll bar
	Spax adjustable spring/damper units



Rear suspension	Upper and lower parallel trailing links, Panhard rod, Spax adjustable spring/damper units
Brakes	MG Midget front discs, rear drums Goodridge steel-braided hoses
Steering	MG Midget rack and pinion
Wheels	Image split-rim
Tyres	5½ × 13 front, 6 × 13 rear Yokohama A008 185/60 VR13 front 205/60 VR13 rear
Cooling	Modified Caterham Seven radiator, repositioned inlet and outlet spigots, side frames removed and replaced by struts front and rear, fully ducted from nose aperture, exhausting into wheelarches, front-mounted electric fan with adjustable thermostatic control and manual override Fabricated alloy header tank in top hose Expansion tank sited in LH side pod and connected between rear of heater rail and cylinder head, no thermostat

DETAIL ENGINE SPECIFICATION

Base engine	Fiat 131 1600 Twin-Cam
Pistons	Mahle Lancia Beta 1.6, +0.4mm, with modified valve pockets, weight-matched
Compression ratio	10.0:1
Cylinder head	Ported and flowed by GCT GC race head bolts
Valves	Standard: 41.8mm inlet, 36.0mm exhaust, fully ported/blueprinted seats
Valve guides	Colsibro
Valve springs	GC triple interference
Camshafts	GC St III (3A)
Cam lift	10.8mm
Cam timing	Timed at 110° FL
Cam drive	1" belt
Cam pulleys	Retimed and dowelled
Crankshaft	Keyholed main journal oil holes Threaded oilway blanking plugs Balanced
Bearing shells	Vandervell VP19 aluminium/tin
Connecting rods	Weight-matched
Flywheel	Lightened to 12mm thickness Balanced with clutch Dowelled to crankshaft
Oil sump	GC Race Mk2 steel fabricated shallow sump with baffles and swinging gates Modified oil pick-up and breather pipes Mobil 1 5W/50 fully synthetic engine oil
Carburation	2 × Weber 40 DCOE 35 on GC manifold Cold-air box fed from NACA duct in bonnet Pipercross filter, custom-built bellcrank linkage
Choke size	34mm
Fuel pump	Facet Silver Top Competition
Ignition	Standard contact breaker and coil Lucas Silicone plug leads
Spark plugs	NGK B9EGV
Exhaust	Custom-built 4-2-1 exhaust manifold by Competition Fabrications of Attleborough, Norfolk Peco race muffler, 50mm inlet/outlet
Filtration	Remote filter head mounted in LH pod Screw-on canister filter
Water pump	2-litre type
Oil cooling	235mm × 13-row Serck oil cooler mounted behind water radiator and ducted from underside of car, exhausting into wheelarches
Alternator	From Datsun Cherry for minimal size
Engine-mounting rubbers	Caterham Seven
Estimated power	150bhp @ 8000rpm
Estimated torque	124lbf ft @ 5500rpm

18/141, 18/142: John Hornsby of Chalgrove built this superb replica of a 1955 Porsche RS550 Spyder while studying as a civil engineering graduate at Oxford's Brookes University. Tubular steel chassis and bodyshell came from Barry Martin, who produced around 30 kits in the late-1980s. Car features 2l Fiat engine driving through VW four-speed transaxle (3.88 final-drive with reversed flywheel). Flywheel is special to accept modified Porsche clutch. Engine has GC head, St III rally cams, 124 1800 pistons, 45s, 4-2-1 exhaust, developing around 174bhp at 7000rpm. Beautiful wheels are original Porsche 356, complementing nicely executed replica of very rare and desirable car.

18/143: John Whalley with his immaculate Stratos replica, about which he writes: "The car is built on a chassis supplied by Hawk Cars and designed by Gerry Hawkridge. The design holds very closely to the original cars excepting that the original used a centre steel tub; the replicas have a chassis with integral roll-cage. The car is then assembled from much the same 'kit of parts' as the factory used, with parts from the Fulvia, X1/9, Dino and 124.

With regard to the fact that you can't paint a Rembrandt, you may only make a likeness, I decided to equip the car with an engine that would not have been overlooked if the car had been built by the factory in the late 80s: the turbocharged 2-litre Twin-Cam. This solution had not been available at the beginning of the Stratos project... Today, using the 8-valve Integrale version of the Twin-Cam, we have available the same power (up to 270bhp) as the [Ferrari-engined] works rally cars, but more importantly, we have much more torque – over 260lbf ft at 3000rpm, compared with the Ferrari's 200lbf ft at 6500rpm. Thus we now have a car that can perform 0–60mph in around 4sec, with a standing quarter-mile time of 11.4sec.

My car has now been driven in anger on over 25 different race circuits across Europe as well as on various international rallies and the occasional sprint." (Photo Mary Harvey)

18/144: Well-known Lancia dealer John Whalley with his Hawkridge Stratos replica on a race circuit in France. Totally prepared by himself, John's car, powered by a 16v Thema Turbo engine, is a familiar sight at motorsport events around the UK. Mary Harvey, of Bicester, Oxon, who took this photograph, commented: "John's car is probably one of the most beautiful examples around."

18/145: Tim Ball airborne in his 1600 Fiat TC Talbot Sunbeam. Engine is similar to Rod Bennett's 1600 Delta, developing around 160bhp. A fast, competitive and well-prepared car. (Photo Tony Large Photographic, Reading)

18/146: Graham Hall competes in Class 9 Autograss with his Radford (Redditch)-based club. Service manager of a horticultural engineering company, Graham built the 'special' himself using 3mm box-section for the cage and chassis and 1mm section for the rest of the frame. Engine is a 2l Fiat married to a Mk 9 Hewland gearbox. One attraction of Autograss is the number of meetings available to racers – Graham does about 20 per year.

18/147, 18/148: Tim Haddon modified a Lancia Monte Carlo to produce this exceptional 037 'lookalike'. Main job was to shorten car by 7"! Spaceframe at rear holds GC St II 2l, approx 176bhp. (This car was featured in the excellent Cars & Car Conversions magazine some years ago.)

18/149: Killarney and District (Eire) MC '95 2l Champion Fergus O'Meara competing with his rallycross Escort-Fiat.

18/150: Colin Haggett's superb Darrian rally car equipped with the Fiat engine examined in Case History No 5.

APPENDIX A

SUGGESTED COMPONENT LIFE SCHEDULE/REPLACEMENT GUIDELINES (COMPETITION ENGINES)

COMPONENT	TURBO + FULL RACE (n/a)	RALLY OR RACE III	ST II	NOTES
CON-RODS	36 HR **	60 HR **	100 HR **	(ENGINE REBUILD)
CON-ROD BOLTS + NUTS	24 HR *	40 HR *	80 HR *	
FLYWHEEL BOLTS	AS ROD BOLTS			
FORGED PISTONS	48 HR **	100 HR **		CRACK TEST BEFORE DISCARDING
ROD BEARINGS				
–VP 2	24 HR **	40 HR **	60 HR **	FIGURES FOR DRY SUMP, FOR WET
–VP 19	12 HR *	20 HR *	30 HR *	SUMP REDUCE HOURS BY 50%
MAIN BEARINGS				
–VP 2 –VP 19	{ 24 HR **	{ 40 HR **	{ 60 HR **	INTERMEDIATE CHECK ONLY IF ROD BEARINGS DAMAGED
RINGS	12 HR *	20 HR *	40 HR *	NOT ESSENTIAL BUT OPTIMUM,
VALVE SEATS & GUIDES	12 HR **	20 HR **	40 HR **	ALSO REFACE VALVES
VALVE SPRINGS	24 HR *	40 HR *	60 HR **	
CRANKSHAFT	12 HR **	20 HR **	40 HR **	
ENGINE OIL & FILTER	12 HR *	20 HR *	AS ST III	DRY SUMP
(NOT CANTON FILTER)	9 HR *	15 HR *	20 HR *	WET SUMP
CAM BELT ¾"	12 HR *	15 HR *	20 HR *	INCREASE CHECKS IN DUSTY / DRY
1"	24 HR *	30 HR *	40 HR *	CONDITIONS (ESP RALLY)
HEAD BOLTS	20 HR *	40 HR *	80 HR *	

NOTES:

* RENEW

** INSPECT AND RECTIFY AS REQUIRED.

– HOURS GIVEN ARE 'RACE HOURS' WHERE 2 RACE HOURS ROUGHLY CORRESPONDS TO 1 RACE OR RALLY EVENT (INCLUDING RELEVANT TESTING AND PRACTICE).

– CHECK ALL EXTERNAL FASTENERS AFTER EVERY EVENT.

– FOR SHOT-PEENED CON-ROD LIFE ADD 20%.

APPENDIX B

INITIAL START-UP (ALL ENGINES)

1 Ensure all installation fasteners and electrical connections are secure, *eg* bellhousing, engine mountings, starter motor, sump plug. Fill engine with oil to level and fill cooling system with plain water. Check for leaks.
2 Check engine is earthed correctly. Connect all gauges. Ensure low-pressure oil light shows when ignition is switched on. Check clutch play.
3 Switch off fuel pump.
4 Remove spark plugs, fill cam boxes with oil and apply cam start-up lubricant (new cams only) and crank up oil pressure. If oil cooler is fitted, split hose lines and fill separately from oil can. With dry-sump systems spin oil pump via electric drill. If oil light does not go out, remove oil filter housing and prime oil pump with engine oil and try again. (If ignition is electronic, temporarily isolate main ignition power feed.)
5 Assuming oil pressure is OK, check for leaks, then replace plugs. Switch on fuel pump (check for leaks).
6 Start engine and adjust tickover speed to 2000rpm. If engine runs very unevenly, likely causes at this stage are carbs out of balance/idle mixtures/ignition timing. Initially check ignition timing with strobe light at 1000rpm and reset distributor as required.
7 Roughly set idle mixture and balance and warm-up engine at 2000rpm, ensuring engine oil and coolant temperatures do not exceed settings and that oil pressure is stable at minimum 40lbf/in^2. When warmed-up, restore tickover to 800–1000rpm.
8 With engine warm, check for leaks of oil/fuel/coolant and carry out full fuel/ignition system setting-up procedure.
9 Check oil level.
10 Engine is now ready for running-in.

APPENDIX C

ENGINE TRACKSIDE DIAGNOSIS FOR SUDDEN POWER LOSS

SYMPTOM	CAUSE/CHECK	CAUTIONARY NOTE
Engine smoking	**A** Suspect detonation – inspect plugs for colour/blistering – check ignition timing – check fuel pump output/jets etc – compression check (hot, plugs out, wide open throttle) – check engine temperature/cooling system – remove head and **inspect for signs of detonation** (important)	Detonation can crack pistons **do not race** if evident
	B Suspect over-rich mixture – check plugs – check fuel pump output/carbs flooding (floats?) – check air filter – check exhaust layout (back pressure) – compression test (if low, suspect ring damage)	If smoking still evident **do not race**
	C Suspect bearing damage – check catchtank – check oil system, level/pressure – remove sump and check	If bearing damage evident **do not race**
Engine not smoking	**D** Suspect valve damage – check cam belt and cam timing – compression test (if compression gauge not available, check valve/shim clearances) **E** Suspect ignition failure/plugs – crank engine and check sparks – no fuel supply (check also A) **F** Suspect blown head gasket – inspect coolant for oil and *vice versa* – compression test	Overhaul required

This is a random summary of some of the trackside faults seen by GCT over the years!
Gauze filter inhaled into carb
Rampipe stud in inlet tract
Cracked pistons
Broken rings
Worn rings/bores
Cylinder liner shifted
Blown head gasket
Cracked head
Bent valve
Broken valve
Broken valve spring
Broken auxiliary driveshaft
Sheared oil pump drive
Broken crank front key (caused by loose pulley)
Damaged bearings
Gearbox input shaft rubbing on crank
Crankcase pressure – blowing sump gasket
Defective ignition amplifier
Broken distributor rotor (not to mention faulty components!)
Seized bobweights
Firing order wrong (HT leads)
Throttle linkage falling apart
Defective fuel pump
Air filter backplate blocking float chamber vents (GC Hydro!)
Cam belt loose
Stones under cam belt (timing slipped)
Carb secondary venturis rotating (should be lock-wired)
Worn out camshaft
Road plugs in race engine (*sic!*)
[The author would be pleased to hear of new and interesting additions to this list!]

APPENDIX D

HINTS ON ROLLING-ROAD TUNING

Writes Gerard Sauer...
No matter how you have done your engine build or tuning work, possibly the best way to check and set the performance of a tuned engine fitted to a car is to run it on the rolling-road. You can imagine the kind of thing that the rolling-road would do, but without sounding condescending, do you know how best to prepare for this and how to avoid disaster?

Disaster? I can hear you ask. Why should there be disaster when the engine is brand new and built to perfect specification?

Well, for a start, it is a definitive no-no to run any brand new engine under full load straight away, and thus the answer lies in the nature of what the rolling-road does and how it tests your engine, and it also depends on all the other components and systems surrounding your engine.

As all this can get quite complicated, for the sake of convenience I have divided the chapter into a number of sections, which relate to the various types of cars and their drivelines as well as the conditions that you may come across.

For example, if you had a 4WD car you would need a 4WD rolling-road, but you also need to take some precautions to avoid damaging the drivetrain. Similarly, if you had just fitted a brand new, never run engine, you would need to prepare for that situation to prevent early damage.

As you will see, preparation is the key word here, but in addition it is also very important to be certain that the rolling-road and its operator are a professional outfit with good and well maintained equipment.

One of the most important pieces of equipment is the air exchange facilities which have been provided for the rolling-road, as this should facilitiate good repeatable runs and safe at-the-limit operation. Big fans are needed: fans that can replace the air in the cell or space several times over during full-power runs. The other piece of equipment that is of vital importance is the exhaust gas extraction arrangement; please don't stand around power runs with the exhaust gas going into atmosphere around you – this can be extremely dangerous. It also makes any measurement irrelevant. In my experience it always pays to get to know the man and his equipment before committing your pride and joy for a workout.

Now, to get down to specifics, I have divided what follows into these six sections:

Section 1: PREPARATION
Section 2: TWO-WHEEL DRIVE, FWD & RWD
Section 3: FOUR-WHEEL DRIVE
Section 4: NEW ENGINES, OLD ENGINES
Section 5: TURBO ENGINES
Section 6: AFTERCARE

Section 1: PREPARATION

Going on the rollers with your car is always a nerve-wracking time, so these few bits of advice are aimed at making life less harrowing and more rewarding.

Probably the best advice that I can give you is to understand that the rolling-road is not a very accurate device, and that therefore you should not attach a great deal of significance to the results.

This may sound strange, but the fact is that rolling-road results are subject to much greater variation than, for example, the dynamometer, where we can control the running conditions of the engine much more tightly. So don't start looking for the smallest gains or differences, but always compare two or three runs with each other.

The secret of using the rolling-road as a measuring tool is all to do with monitoring the conditions of the measurements. For example, if you have just done a run with the engine at, say, 5000rpm on full load, and then you interrupt the testing for, say, 15–20 minutes in order to put something right, but fail to make any adjustment, then start the engine up and immediately take a reading, you might well see an increase in power on the display, but this will almost certainly be false, because the gearbox oil will be hot, but the engine will have cooled down a little and the charge air temperature dropped with it and you might have gained 3–4bhp. The message is that you should always be prepared to have control of the conditions of the test; all manner of things, including the ambient temperature, should be monitored. So, assuming you are ready to go on the rollers with your car, what should you do to make sure your engine does not come to grief, and the results can be trusted?

Start with the general condition of the electrics and the HT systems, and make sure that you have a set of good spark plugs. Make certain that they have the correct heat range. Oil and filter should also be either new or in good condition.

Be sure that the tyres are in good condition and have the same circumference. Also make sure that the tyre pressures are set to specification, or even a little higher, for the test. Make certain that you have enough antifreeze in the system as this raises the boiling point of the coolant. Check that the hoses are in good condition, also that the throttle cable gives full throttle, and set the tappets if appropriate.

Ensure that the car is fitted with accurate instruments, especially for the coolant temperature and oil press/temp. Set the ignition timing. Set the CO and the idle speed. Make sure the car's wheels are in good condition, and if slicks (or 'wets') are fitted, replace them with road tyres (race tyres will rapidly 'self-destruct' on a rolling-road due to overheating). You should also check the exhaust system for leaks. Finally, make sure there is enough oil in the gearbox and the diff units.

If the car has an undertray, remove it so that enough air can circulate around the whole engine and gearbox assembly. The rolling-road operator will be able to tell you what sort of horsepower the road can take, and it is best not to go to a rolling-road that is working in excess or near its capacity as this can make measuring and comparing very difficult.

The next thing to worry about is the rolling-road itself, what it consists of and what level of equipment you need. I say what *you* need because, if you are just doing a quick check to see why there is a misfire or a hesitation, you will need different equipment from that used to get the fuelling right. You will need an oscilloscope, an infra-red gas analyzer for CO, HC meter and, preferably, a Lambda sensor.

All this equipment should be available at a good rolling-road; if not, it is time to be asking some questions. Remember that it is your engine and car, and if the rolling-road operator does not have the right gear you may not be aware of the damage being done before it is too late.

Doing things right sometimes requires a bit of planning and forethought. You may want to do some experimentation by changing some of the components on the engine such as, for example, the inlet trumpet length, or a change of air-filter or ignition system. If so, prepare everything beforehand; there is no sense is having the use of the rolling-road for a day only to waste most of it bolting up things and

modifying parts to fit onto the engine.

Make sure that everything you want to try has been pre-fitted and assembled at least once to prove that it works and operates properly. Sit down and think about what you want to do, write down the order in which you want to try things and give a copy to the operator so he will understand what you want and, more important, why, and the day will be so much more rewarding.

Another aspect of preparation is to take with you any bit of information you have on your engine, and any special tooling or bits that make life easy to adjust or check the unit. Also take a pen and A4 pad and have someone with you to help write things down whilst you make the changes so that you have a record of them.

Try also to obtain a reading of ambient temperature, humidity and barometric pressure. You can usually get these values from the local weather station or nearby airfield. This information will allow you to compare rolling-road sessions with one another as you can make adjustments for the conditions on the day.

You are now ready to face the rollers, but remember, *not* with a new engine; before going on the rollers with a new engine, read *Section 4*.

Section 2: TWO-WHEEL DRIVE – FWD & RWD

There are some bits of basic advice that we have covered in preparation which apply equally to all engines and vehicles to be tested. But here we are looking at matters specific to front-wheel-drive or rear-wheel-drive vehicles.

Front-wheel drive

Going on the rollers with front-wheel drive you will have the following to think about and act upon:

The first task is to tie the car securely onto the rollers, and with FWD cars this is more of a problem than with RWD cars. Both need to be secured, but the FWD car is more prone to side-to-side wandering because the steering wheels are also the driving wheels, and therefore the car will need to be strapped down and secured side-to-side as well. This is usually done with the help of sturdy straps, rather like safety belts, but it can also be done with the help of wheel supports fitted to the side of the front wheels, which prevent sideways movement of both wheels. It is also important to tie the car down well enough, otherwise it may want to climb out of the rollers. Tying down properly also prevents wheelspin, and therefore contributes to achieving accurate readings.

Most rolling-roads have anchor points set into the concrete floor from which the car is held in place firmly. The thing to watch for is that the straps do not damage your bodywork or spoilers and that the front of the car does not sit on the concrete; remember, the wheels will sink deep into the rollers once the operator has lowered the car onto them ready for action. Special care, therefore, is needed in this respect if the car in question has already had its ride height lowered. You can easily overcome this by making some wooden sandwich pieces that fit into the spring seats and temporarily raise the car to allow the necessary clearance.

Another reason for raising lowered cars is to improve the flow of air under the car to cool the transmission. When running under full-throttle conditions it is vital that cool air is blown underneath the car. Also ensure that the cooling fan is partly directed into the fresh air intake, or otherwise that a hose or duct is placed in front that will direct the cool air into the intake. Make sure that the exhaust is not grounding and that there is no oil on the ground to get ignited as some exhaust parts will get red hot under full-power runs. Remove the undertray as well.

Rear-wheel drive

Here the tying-down procedure is just as important, but the front wheels have no influence as they are not in the rollers, so the sideways movement of the car under power is not so strong. However, the rear wheels of a conventional RWD car are usually the lightest pair, and you might want to be prepared by taking some empty jerrycans that can be filled with water and put in the boot to stop wheel-spin from spoiling the runs. Just as with the FWD cars, there should be enough clearance underneath the car for cool air to pass under the transmission. Usually the operator will warm-up the transmission before any power runs are taken, and this goes equally for both FWD and RWD cars. Again, lowered cars can cause a problem, and raising them can be done as indicated before. Don't forget to take the shims or spacers out afterwards!

Section 3: FOUR-WHEEL DRIVE

The 4WD cars have the hardest time on the rollers as the transmission can get very hot and the cooling fans are usually able to cool only part of the underneath. See if you can find a rolling-road with under-car cooling fans. There are a few additional things to bear in mind if you are rolling-road-testing a 4WD car.

The first is that it is more important than ever to make sure the transmission temperatures are monitored closely since this will influence the readings that you are obtaining. Some form of surface temperature measurement would help to ensure that the runs are all done under equal conditions. Secondly, there are two sets of rollers, and the pair that is used to adjust for the car's wheelbase needs to be set to the correct space to ensure that both sets of wheels are sitting equally in the rollers. If one or the other set is too far out of the centre the readings will not be very consistent. It is more important than ever to have the tyre pressures and the circumferences checked beforehand. If there is a lot of difference between them, the resultant speed differential and scrub will ruin good measurement and accuracy.

Any 4WD car that has automatic driveline control should be checked over first, and best of all get in touch with the makers as it is possible that the system will be damaged by using a rolling-road. At the front, the tying-down should be carried out as with FWD cars to prevent sideways movement under load.

Section 4: NEW ENGINES, OLD ENGINES

New engines

As with all things in life, nothing lasts forever, but there is a lot you can do to make your engine last a long time and be healthy in the process. The secret is in the running-in process and how you subsequently set up the engine for full power. So let's look at the scenario when you have just completely rebuilt your engine, and particularly those instances where you have made some changes to the specification or done some more radical modifications. For example, the fuelling and the ignition settings may have to be reset to suit the new spec.

This is invariably the most difficult thing to deal with as you cannot run the car on the rollers under full load, and the settings will not be very accurate. The biggest problem to overcome is the possible over-fuelling of the engine, as this will wash the oil off the bore walls and damage the valve guides in very short order. The solution is to err on the side of caution. You can go on the rollers if you want to, but you should avoid full-power runs and strain and stress on the engine. For example, if you have just fitted a new cylinder head of higher compression to the engine, set the timing correctly during running-in so that there can be no detonation. Be cautious with fuelling, and ask GCT to give you an idea of the settings that are going to get you safely

through the running-in period.

Assuming that you have completed the running-in period and you are about to go onto the rollers, the first thing to do is to get the fuelling right. The operator should take short runs and begin at the lower end of the scale first, carefully watching the temperatures and listening all the time for detonation noises. Make certain that you have a good, reliable way of measuring the mixture. The best I use is a self-contained Lambda sensor from America made by NTK, but in the UK there is a similar device on sale from Weber Concessionaires through their dealer network or Lumenition.

This sensor, when fitted into the exhaust system, gives an almost instant readout of the mixture air-to-fuel ratio and prevents you having to hang on long during power runs to ensure that the infra-red analyzer can capture enough exhaust gas to give you a sample strength. As the Lambda sensor is fitted near the front of the exhaust, it is also less prone to false measurements due to air leaks in the exhaust system. Using this method a reasonable mixture can be set quickly and accurately.

By now you will be able to see whether you have decent power or not, as the case may be. If you are very short of what the engine should be giving, don't keep thrashing the unit or blaming the operator, but investigate first. Check cylinder leakdown figures and compressions: a good engine built to fine tolerances should have leakdown figures around the 92–94% average; anything lower than 80% is unacceptable.

Don't be too despondent, though; the remedy is to get the car home and investigate the cause. Once everything is set up properly, and given that the engine is good, do a final confirmation run to give a consistent powergraph, take your readings, do the temperature measurements and then call it a day.

Old engines

There is no hard and fast way of knowing what you can expect from an older engine, but if the leakdown figures are still good and there is not too much engine breathing, the figures should be within 10–12% of new. You will have to be much more careful at the top of the power curve with high revs, and be sure to give the engine some new oil beforehand.

Things like oil and coolant temperatures should be watched much more closely, and if you are going to set the ignition timing, be careful not to overdo the advance as this can cause damage much more quickly than on a new unit, particularly with engines that have a fair amount of coke build-up in the combustion chamber. Be aware also that the cooling system may be silted or the head gasket might be weak, and of course that the rolling-road imposes much more strain than the open road.

Section 5: TURBO ENGINES

In the case of the naturally aspirated engine, most areas of concern have been covered, but in the case of turbocharged or supercharged engines it is worth making a few observations that should help you avoid the more obvious problems. When I first rolling-road-tested a V6 twin-turbo Capri 2.8 engine in the early 1980s, I received an immediate lesson in how not to do it. We were taking our first full-power run at around 4500rpm when there was a flash from under the bonnet and an instant raging fire ensued. This, once extinguished, turned out to have been caused by the radiated heat from the turbo melting the plastic oil pressure pipe on the side fitted to the inner wing panel, some 6 inches away from the manifold.

The fact is that with the engine on full boost it is not uncommon – and quite healthy – to find that the whole of the exhaust manifold is white hot and radiating enormous amounts of heat. So the first thing to look for is a good working fire-extinguisher or two. Don't be afraid to demand that they are kept nearby. It is your car, your engine and your hard work.

In the case of turbos, it is even more important to undertake only short runs, say no more than three at the time lasting about 8–10 seconds steady each, then a break of approximately 10–15 minutes, then to repeat the process at a further three intervals of 500rpm apart. This way you can avoid damage and prevent the temperatures rising too far and making the measurements worthless. Be certain to have good shielding of the exhaust components, and keep an eye on the floorpan as well; I have seen the underfelt catch fire inside the car! Make sure that there is fresh air onto the turbo, and I sometimes use one of the exhaust extractors that hang from the roof to suck air over the turbo and circulate cool air that way around the unit.

Oil temperatures must be watched, and the transmission also gets a lot more stick and tends to run hotter. Allow the engine to run unloaded between runs for a while to prevent sudden drops and rises in temperature, which can cause distress fractures in your manifold or turbo housing. If you can, you must monitor the charge air temperature as this will give you accurate results and measurements; if possible, measure intercooler in and out temperatures to allow precise conditioning.

The typical inlet temperature that you are looking for under maximum boost is 45–50deg C. Any more than 80–85deg and the readings will be less relevant and the chances of detonation will grow. Be certain to adjust the ignition timing off-boost first, then set the boost retard if you can, but keep watching those numbers on the temperature gauges and the Lambda sensor or the gas analyzer. Anything below 4.5% CO is dangerous, and an AFR in excess of Lambda = 1, *ie* 14.5% or above under full load, must be regarded with the utmost suspicion.

Start from the slightly over-fuelled position then reduce to suit. Remember that too much fuel in the off-boost regions makes for poor response from the turbo and lengthens the lag period. Any consistent over-fuelling also causes fuel contamination of the oil and reduces engine life dramatically. There are many racers that use the extra fuel as a cooling agent on the pistons, but for the road this is to be avoided at all cost, as the only result is an engine rebuild every 5000 miles or so.

Run the engine with the breather system venting to air, not into the inlet tract, but if you have to, install an oil-air separator first before going on the rolling-road. Take a plug reading if you are not sure how close you are to the limit with the ignition advance, but I always use a magnifying glass to have a close look. Advancing the engine makes horsepower, but going too far makes nothing but holes everywhere, so be cautious.

Section 6: AFTERCARE

There is nothing quite so good as to come away from the rolling-road and to give your car a workout with everything functioning to its best possible settings; the engine will respond better and the power should be noticeably up from its previous performance.

First change the oil again, and the filter. Set the idle speed and the CO. Inspect everything around the engine for leaks and check the transmission for same. Remove any test equipment that is not permanent. Remove the lowering spacers. Write down all the settings, including jets sizes, plug spec and gaps. If you have raised the tyre pressures, bring them back to standard. Any notes you have made about things that need to be changed or improved, do them now, don't wait until later.

APPENDIX E

TYPICAL ENGINE SPECIFICATION SHEET

1 COMPLETE ENGINE/BLOCK/HEAD/OTHER: ____________________
2 DATE: ________________ 3 OWNER: ____________________
4 TEL NO: ________________ 5 ADDRESS: ____________________
____
6 CAR/TYPE: ________________ 7 STAGE: ____________________
8 BORE: __________mm 9 STROKE: __________mm 10 CAP: __________cc
11 PISTONS: ________________ 12 HEAD VOL: ________________cc
13 CR: ________________ 14 FUEL: ____________________
15 CRANKSHAFT PREP: ________________WT: ____________________
16 JOURNAL SIZES: M ________________B ____________________
17 BEARING TYPES: M ________SIZE ________B ________SIZE __________
18 THRUST WASHERS: TYPE __________SIZE________END FLOAT__________
19 ROD PREP: ________________WT: ____________________
20 FLYWHEEL PREP: ________________WT: ____________________
21 CLUTCH TYPE: ________________SPLINE: ____________________
22 BALANCING: ____
23 VALVES: INLET SIZE_______TYPE________EX SIZE ______TYPE ________
24 VALVE SEATS: INLET ________EX ________ GUIDE: TYPE ________
25 VALVE SPRINGS: ________________SET-UP HT ______________mm
26 VALVE CAPS: ____
27 CYL HEAD MODS: ____
________________GASKET: ____________________
28 BLOCK MODS: ____
____
____
29 INLET MANIFOLD/FUEL SYSTEM: ____________________
____
____
30 LINKAGE: ____
31 AIR FILTER SYSTEM (MANDATORY): ____________________
32 CAMSHAFTS: INLET TIMING ___________LIFT (NOMINAL) ________mm
PROFILE NO ________EX TIMING ________LIFT(NOMINAL) ________mm
33 CAM WHEELS: TYPE ________________BELT: ________________
34 VALVE CLEARANCES (COLD) IN ________________EX ________________
35 FUEL PUMP: ____
36 SPARK PLUGS: ROAD __________RALLY __________RACE __________
GAP____________________TORQUE: 12lbf ft
37 IGNITION SYSTEM: ________________TIMING________________
38 ENGINE OVER-REV LIMITER (MANDATORY)____________________
39 LUBRICATION SYSTEM: ____________________________________
________________________CATCH TANK: ____________________
40 OIL COOLER: ____
41 COOLING SYSTEM: ____
42 EXHAUST SYSTEM: PRIMARY PIPE LENGTH ____________ID ________
TYPE 4-2-1/4-1 SECONDARY PIPE LENGTH ____________ID ________
TAIL PIPE (4-2-1 ONLY) LENGTH ____________ID ________
SILENCER TYPE ____________________
43 FASTENERS: HEAD BOLTS ______________TORQUE______________
CON ROD ______________TORQUE______________
FLYWHEEL ______________TORQUE______________
OTHER____
HEAD BOLTS RE-TORQUE REQD YES/NO

44 OIL: RUNNING IN: SEMI SYNTHETIC OR MINERAL CLASS API SH, CCMC G5, 10W/40 OR BETTER. POST RUN-IN: (*SEE CHAPTER 13*).

45 WORKING TEMPERATURES: COOLANT PREF 70–75°C MAX 80°C, OIL 85°C ± 10°C.

46 OIL PRESSURE: (AT WORKING TEMPERATURE) WET SUMP – MUST GIVE 15–25lbf/in^2 AT TICKOVER (800–1000rpm) WITH MIN 40lbf/in^2 UNDER LOAD (FROM 2000rpm UPWARDS) INCREASING BY APPROX 10lbf/in^2 PER 1000rpm TO A MAXIMUM OF BETWEEN 70–100 DEPENDING ON ENGINE AND INSTALLATION (LENGTH OF COOLER LINES, SIZE OF COOLER, PUMP TYPE ETC). OIL TEMPERATURE IN EXCESS OF RECOMMENDATION – PARA 45 – WILL DRASTICALLY REDUCE OIL PRESSURE AND LEAD TO BEARING FAILURE. DRY SUMP – AS WET SUMP BUT WITH LOWER (55–79lbf/in^2) MAX PRESS AND GREATER DELIVERY.

USEFUL CONTACTS

Guy Croft Race Engines
e-mail: gcengines@aol.com
www.Guy-Croft.com
Address: PO Box 699
Wellingore Lincoln UK
LN5 0XD

VEHICLE AND SPARES SPECIALISTS
Middle Barton Garage (specialists in Fiat and Abarth parts and services, models 1955–1996), North Street, Middle Barton, Oxfordshire OX7 7BH. Tel: 01869 340289; fax: 01869 340110. Contact: Tony Castle-Miller

Richard Thorne Classic Cars (Lancia specialists, especially Integrale), Unit 1 Bloomfield Hatch, Mortimer, Reading, Berkshire RG7 3AD. Tel: 01734 333633; fax: 01734 333715. Contact: Richard Thorne

Martin Sismey Fiat/Lancia Supplies (specialists in sourcing new and used car/engine parts and donor engines), 5 Thorneymill Road, Ritchings Park, Iver, Buckinghamshire SL0 9AQ. Tel/fax: 01753 654102. Contact: Martin Sismey

Scorpion Reproductions (extensive range of obsolete and reproduction parts for TC vehicles, especially Beta and Montecarlo, restorations), Llangorra, Wheal Butson, St Agnes, Cornwall TR5 0PU. Tel/fax: 01872 553272. Contact: Heather Pastor

FUEL SYSTEMS
Datum (Weber, Dellorto suppliers), 180 Hersham Road, Walton-on-Thames, Surrey KT12 5QE. Tel: 01932 221955; fax: 01932 246859. Contact: Terry Huxley

Fuel System Enterprises, 180 Hersham Road, Walton-on-Thames, Surrey KT12 5QE. Tel: 01932 231973; fax: 01932 246859. Contact: Peter Huxley

Autocar Electrical Equipment Co Ltd (Lumenition FI and rev-limiters), 49-51 Tiverton Street, London SE1 6NE. Tel: 0171 403 4334; fax: 0171 378 1270. Contact: Chris Bailey

AIR FILTERS/RAMPIPES
Pipercross, Filtration House, Overstone Road, Moulton, Northampton NN3 1UL. Tel: 01604 671100; fax: 01604 671101. Contact: Phil Davies

ITG Ltd, Unit 5 Fairfield Court, Wheler Road, Seven Stars Industrial Estate, Whitley, Coventry CV3 4LJ. Tel: 01203 305386; fax: 01203 307999. Contact: Jonathan Douglas

EXHAUST SYSTEMS (CUSTOM-MADE)
'Mike the Pipe', 128 Stanley Park Road, Wallington, Surrey SM5 3JG. Tel: 0181 669 1719; fax: 0181 773 4096. Contact: Mike Randall

Maniflow, 64-66 St Paul's Road, Salisbury, Wiltshire SP2 7BD. Tel: 01722 335378; fax: 01722 320834.

PD Gough & Associates (stainless steel specialists), The Old Foundry, Common Lane, Watnall, Nottingham NG16 1HD. Tel: 0115 9382241; fax: 0115 9459162. Contact: Tony Byard

DYNO TESTING
Warrior Automotive Research Ltd, Bellbrook Industrial Estate, Uckfield, East Sussex TN22 1QL. Tel: 01825 764833; fax: 01825 769132. Contact: Russell Pain

INSTRUMENTS
Raceparts UK Ltd, Unit 3 Rockfort Industrial Estate, Wallingford, Oxfordshire OX10 9DA. Tel: 01491 837142; fax: 01491 836689.

Stack Ltd, Unit 10 Wedgwood Road, Bicester, Oxfordshire OX6 7UL. Tel: 01869 240404; fax: 01869 245500.

OIL COOLERS AND SYSTEMS
Think Automotive Ltd (Mocal products, Aeroquip), 292 Worton Road, Isleworth, Middlesex TW7 6EL. Tel: 0181 568 1172; fax: 0181 847 5338.

Procomp Engineering (heat exchangers), 6 The Parklands, Erdington, Birmingham B23 6LA. Tel/fax: 0121 350 3528.

MAPPED IGNITION SYSTEMS
MBE Systems, The Elliott Centre, Love Lane Industrial Estate, Cirencester, Gloucestershire GL7 1YS. Tel: 01285 641095; fax: 01285 641096. Contact: Jeff Moore

TURBO SYSTEMS, MAPPING
Greenlight Tuning Ltd, 31 Padfield Road, London SE5 9AA. Tel: 0171 733 7024; fax: 0171 737 1769. Contact: Gerard Sauer

DRY-SUMP PUMPS, PANS
Titan Motorsport, The Harley Works, Paxton Hill, St Neots, Cambridgeshire PE19 4RA. Tel: 01480 474402; fax: 01480 405668. Contact: Oz Timms

TOOLS
Pacehigh Ltd (Flex-Hone), PO Box 128, Hatfield, Hertfordshire AL9 5LB. Tel: 01707 665707; fax: 01707 276919.

Morrisflex Ltd (die grinders and shafts), London Road, Braunston, Northamptonshire NN11 7HX. Tel: 01788 891777; fax: 01788 891629.

Garryson-Insley Ltd (burrs, abrasives), Spring Road, Ibstock, Leicestershire LE67 6LR. Tel: 01530 261145; fax: 01530 262801.

ATA Grinding Processes Ltd (die grinders, burrs and abrasives), ATA House, Boundary Way, Hemel Hempstead, Hertfordshire HP2 7SS. Tel: 01442 64422; fax: 01442 231383.

Medway Tools (Metabo die grinders and shafts), Unit 4 Keel Court, Enterprise Close, Medway City Estate, Rochester, Kent ME2 4LY (adjacent to GCT). Tel: 01634 290567; fax: 01634 720003.

GEARBOXES, DIFFERENTIALS
Tran-X Gears, Bartleet Road, West Washford, Redditch, Worcestershire B98 0DG. Tel: 01527 510720; fax: 01527 510435. Contact: Dave Hirons

CATCH TANKS, DRY-SUMP TANKS
Brise Alloy Fabrications, rear of Motoplat, 117 Dartford Road, Dartford, Kent DA1 3EN. Tel: 01322 222343; fax: 01322 289935. Contact: Tim Honess

PLATING
Corrotect Electroplating, Pleasant Row, Brompton, Gillingham, Kent ME7 5QY. Tel: 01634 8456636; fax: 01634 832008. Contact: Cyril Ellis

CLUBS
(Well worth joining - both clubs have extensive 'cars' and 'spares for sale' sections in their magazines and publish regular technical features and details of events. Note: Contacts are volunteers, so please bear this in mind when phoning and enclose an SAE when writing!)

Fiat Twin-Cam Register, Graham Morrish (Membership Secretary), 19 Oakley Wood Road, Bishops Tachbrook, Leamington Spa, Warwickshire CV33 9RW. Tel: 01926 335097.

Lancia Motor Club, David Baker, Mount Pleasant, Penhros Brymbo, Wrexham, Clwyd LL11 5LY. Tel (evenings only): 01978 750631.